JOURNAL of the AMERICAN SOCIETY *of* *NAVAL ENGINEERS* Inc.

August, 1956

Volume 68 Number 3

Presidents of the
AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.

Rear Admiral Kenneth K. Cowart, U.S.C.G.

Past Presidents

1889-90 Chief Engineer *Nathan P. Towne*, U. S. Navy
1891 Past Assistant Engineer *G. W. Baird*, U. S. Navy
1892 Chief Engineer *David Smith*, U. S. Navy
1893-94 Chief Engineer *H. Webster*, U. S. Navy
1895 Chief Engineer *James H. Perry*, U. S. Navy
1896 Chief Engineer *E. D. Robie*, U. S. Navy
1897 Chief Engineer *David Smith*, U. S. Navy
1898-99 Chief Engineer *H. Webster*, U. S. Navy
1900 Commander *H. Webster*, U. S. Navy
1901-02 Commander *C. W. Rae*, U. S. Navy
1903 Lieutenant Commander *W. M. Parks*, U. S. Navy
1904 Rear Admiral *John D. Ford*, U. S. Navy
1905 Commander *A. F. Dixon*, U. S. Navy
1906 Commander *A. B. Canaga*, U. S. Navy
1907 Commander *B. C. Bryan*, U. S. Navy
1908 Commander *R. S. Griffin*, U. S. Navy
1909 Rear Admiral *John K. Barton*, U. S. Navy
1910-11 Engineer-in-Chief *H. I. Cone*, U. S. Navy
1912-13 Captain *R. S. Griffin*, U. S. Navy
1914 Rear Admiral *J. R. Edwards*, U. S. Navy
1915 Captain *S. S. Robison*, U. S. Navy
1916 Captain *C. W. Dyson*, U. S. Navy
1917-18 Captain *H. P. Norton*, U. S. Navy
1919 Rear Admiral *H. P. Norton*, U. S. Navy
1920 Rear Admiral *C. W. Dyson*, U. S. Navy
1921 Captain **A. J. Hepburn**, U. S. Navy
1922 Rear Admiral *C. W. Dyson*, U. S. Navy
1923-24 Captain *J. T. Tompkins*, U. S. Navy
1925 Rear Admiral *C. F. Hughes*, U. S. Navy
1926 Rear Admiral *M. M. Taylor*, U. S. Navy
1927 Rear Admiral *John Halligan*, U. S. Navy
1928 Captain *E. L. Bennett*, U. S. Navy
1929-30 Rear Admiral **H. E. Yarnall**, U. S. Navy
1931 Rear Admiral **S. M. Robinson**, U .S. Navy
1932 Rear Admiral *F. B. Upham*, U. S. Navy
1933 Rear Admiral *Wm. A. Moffett*, U. S. Navy
1933 Rear Admiral *E. J. King*. U. S. Navy
1934-36 Rear Admiral **William H. Standley**, U. S. Navy
1937 Rear Admiral **Alfred W. Johnson**, U. S. Navy
1938 Rear Admiral **Harold G. Bowen**, U. S. Navy
1939 Rear Admiral **S. M. Robinson**, U. S. Navy
1940 Rear Admiral **Henry Williams**, U. S. Navy
1941 Captain *Claud A. Jones*, U. S. Navy
1942 Rear Admiral **Herbert S. Howard**, U. S. Navy
1943 Rear Admiral **J. J. Broshek**, U. S. Navy
1944 Rear Admiral *C. L. Brand*, U. S. Navy
1945 Rear Admiral **E. W. Mills**, U. S. Navy
1946 Rear Admiral **Harvey F. Johnson**, U.S.C.G.
1947 Rear Admiral **C. W. Styer**, U. S. Navy
1948 Rear Admiral **Roger W. Paine**, U. S. Navy
1948 Rear Admiral **Harvey F. Johnson**, U.S.C.G., Ret.
1949 Rear Admiral **T. A. Solberg**, U. S. Navy
1950 Rear Admiral **F. E. Haeberle**, U. S. Navy
1951 Rear Admiral **David H. Clark**, U. S. Navy
1951 Rear Admiral **Louis Dreller**, U. S. Navy, Ret.
1952 Rear Admiral **T. C. Lonnquest**, U. S. Navy
1953 Rear Admiral **Homer N. Wallin**, U. S. Navy
1954 Rear Admiral **Evander W. Sylvester**, U. S. Navy
1955 Rear Admiral **W. D. Leggett, Jr.**, U. S. Navy
1956 Rear Admiral **F. R. Furth**, U. S. Navy

---

*Names in italics deceased

JOURNAL

of

The American Society of Naval Engineers, Inc.

PUBLISHED QUARTERLY

UNDER THE SUPERVISION OF THE COUNCIL

Rear Admiral KENNETH K. COWART, U.S.C.G., President

Rear Admiral RAWSON BENNETT, U.S.N.

Captain PETER V. COLMAR, U.S.C.G.

Rear Admiral R. E. CRONIN, U.S.N.

Rear Admiral L. V. HONSINGER, U.S.N.

Captain PETER HORN, U.S.N.

Commander E. G. KINTNER, U.S.N.R.

MR. JAMES S. MELTON

Lieut. Comdr. J. W. SAWYER, U.S.N.R.

MR. THOMAS H. SHEPARD, JR.

Captain J. E. HAMILTON, U.S.N. (Ret.), Secretary-Treasurer

SIXTY-EIGHTH YEAR

VOL. 68 AUGUST, 1956 NO. 3

Published Quarterly at Washington, D.C., by The American Society of Naval Engineers, Inc.

Entered as Second Class Matter at the Post Office at Washington, D.C., and at Nashville, Tenn. Acceptance for mailing at special rate of postage provided for in Section 1103, Act of October 3, 1917, authorized on July 3, 1918.

Subscription Rates for Non-Members

The subscription price of the JOURNAL, postpaid to points in the United States and possessions is $9.00. Single copies, $2.25. To other countries, $20.00. Single copies, $5.00. All subscriptions payable in advance.

Make checks, drafts and postal money orders payable to THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.

Advertising rates will be furnished on application.

Addresses will be changed at any time, but only upon notification. Immediate notice should be given of any delay in the receipt of the JOURNAL.

Address all communications for the Society to

THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.

Suite 1004, Continental Building
1012 14th Street, N.W.
Washington 5, D.C.

The American Society of Naval Engineers, Inc., has no official connection with the Navy Department.

TABLE OF CONTENTS

PAGE

Secretary's Notes 417

Managing a Revolution
Rear Admiral A. G. MUMMA, USN 425

The Aircraft Carrier in the German Navy
WILHELM HADELER 431

Metallurgy In Atomic Power
Rear Admiral HYMAN G. RICKOVER 441

Technical Progress in Marine Engineering During 1955
The Shipbuilder and Marine Engine Builder, January 1956..... 446

Nuclear Reactor Design Theory
GEORGE LEPPERT 459

Technical Progress in Shipbuilding During 1955
The Shipbuilder and Marine Engine Builder, January 1956 475

Temperature-Induced Stresses in Beams and Ships
NORMAN H. JASPER 485

How Mechanical Engineers Are Educated In Russia
Engineering, February 10, 1956 498

History of Naval Searchlights
ERNEST BOGHOSIAN 503

The Noise Problem on Board Ship
The Marine Engineers and Naval Architect, January, February 1956.. 506

The Alignment of Main Propulsion Shaft Bearings In Ships
J. J. FRANCIS, LT. R. E. KOSIBA and R. A. WOOLLACOTT..... 515

A Scheme for Improvement In Fleet Support
Commander J. H. GARRETT 527

An Interpretation of Research and Experience In Structural Welding
The Welding Journal, March 1956 533

The Fireman Aboard Modern Combatant Ships
GEORGE J. RASCHER 547

Design Considerations For Naval Gas Turbines
G. L. GRAVES, JR. 554

The Preliminary Design of Surface Warships
ROBERT L. KELLY 565

33,000 Hours On Marine Gas Turbines In Naval Service
Com. W. M. M. FOWLER, JR., and J. W. SAWYER..... 568

PAGE

Reviewing Drawings For Better Ships
ARNOLD BUCHSBAUM 577

Applications of Plastics In Marine Cathodic Protection Systems
H. S. PREISER and M. STANDER 581

The Variation of Temperature In Fixed Fuel Elements of a Nuclear Reactor Under Load and Coolant-Temperatures
WILLIAM J. ROBERTS and IRVING GRANET 591

Use of Sodium and of Sodium-Potassium Alloy as a Heat-Transfer Medium
The Engineer, January 20, 27, 1956 595

Engineering Abstracts
JOHN W. SAWYER 605

Discussion of "Half a Century of Progress In Boiler Water Treatment"
CAPTAIN HENRY C. DINGER 609

Book Reviews and Notices 615

DEPARTMENTS

Obituary 619

Deaths 620

Changes in Membership 621

Association Notes 623

Editor: Captain J. E. HAMILTON, USN (Ret.)
Assistant Editor: Captain ROBERT B. MADDEN, USN
Administrative Assistant: Mr. A. G. FESSENDEN

INDEX to ADVERTISERS

Name	Manufacturers of or Dealers in	Post Office Address	Page
A			
Allis Chalmers Mfg. Co.	Ship Propulsion and other Equipment for Marine Use	Milwaukee, Wisc.	iii
American Brass Co.	Engineering Service on Anaconda Metals	Waterbury, Conn.	viii
B			
Babcock & Wilcox Co.	Marine Water-Tube Boilers, Superheaters, Air-heaters, Economizers, Oil Burners, Refractories, Seamless and Welded Tubing	Room 665, 161 East 42nd St., New York 17 N.Y.	x
Bath Iron Works	Shipbuilders and Engineers	Bath, Maine	ii
Bethlehem Steel Co.: Shipbuilding Division	Steam Turbines, Marine Boilers, Oil Burning Equipment, Shipbuilders and Repairs	25 Broadway, New York, N.Y.	vi
C			
Cleveland Diesel Engine Division General Motors Corp.	Diesel Engines	Cleveland, Ohio	xxiii
Collins Radio Company	Designers and Manufacturers of Radio Communication and Navigation Equipment for the Armed Services	Cedar Rapids, Iowa	xvi
Combustion Engineering Co.	Marine Steam Generators, Bent Tube and Straight Tube Boilers, Forced Circulation Boilers, Superheaters, Economizers and Air Heaters	200 Madison Ave., New York City, N.Y.	xv
Cooper-Bessemer Corp., The	Marine Diesel Engines	25 West 43rd St., New York, N.Y.	xxii
Crane Co.	Crane Valves	836 S. Michigan Ave., Chicago, Ill.	xiii
Cutler-Hammer, Inc.	Electric Control Apparatus	1354 St. Paul Ave., Milwaukee, Wisc.	xiii
D			
DeLaval Steam Turbine Co.	Steam Turbines	485 Nottingham Way, Trenton, N.J.	xvii
F			
Foster-Wheeler Corp.	World's Largest Floating Power Plant	165 Broadway, New York, N.Y.	xi
G			
General Electric Co.	Electric Machinery of all classes	Schenectady, N.Y.	xxi
Gibbs & Cox Inc.	Naval Architects and Engineers	New York, N.Y.	xiii
I			
International Nickel Co.	Producers of Nickel	67 Wall St., New York, N.Y.	iv
J			
Johns-Manville	Insulation, Refractory Cements, Packings, Brake Linings, Asbestos Ebony for Switch and Panel Boards, Waterproofing, Transite Asbestos Sheets, Transite Pipe, Roofing and Shingles, Flooring Insulating Board Products	22 East 40th St., New York, N.Y.	xii
K			
Kaiser Aluminum & Chemical Corp.	Aluminum Products and Engineering Services	Palmolive Bldg., Chicago, Ill.	xiv
Kingsbury Machine Works, Inc.	Thrust Bearings, Journal Bearings, Thrust Meters	Frankford, Philadelphia, Pa.	xxiii
L			
Lukens Steel	All Purpose Steel, "T-1" complete line of Alloy Steels	Coatesville, Pa.	xx

Name	Manufacturers of or Dealers in	Post Office Address	Page
M			
Moffitt, Lucian Q., Inc.	Cutless Bearings for Stern Tubes and Struts	Akron, Ohio	xvi
N			
Newport News Shipbuilding and Drydock Co.	Naval and Merchant Vessels	Newport News, Va.	vii
R			
Radio Corporation of America	Radio Equipment	Camden, N.J.	v
Rosenblatt, M. & Son	Naval Architects, Marine Engineers	253 Broadway, New York, N.Y.	xii
S			
Sperry Gyroscope Co., Inc.	Radar, Loran, Gyro-Pilots, Gyro-Compasses, Electric and Hydraulic Steering Equipments, Magnetic Compass Pilot, Rudder Angle Indicator. Naval Fire Control Instruments, Aircraft Gyropilots, Flight Instruments	Great Neck, L.I., N.Y.	ix
T			
Terry Steam Turbine Co.	Steam Turbines	Hartford, Conn.	xii
U			
U. S. Naval Institute	U. S. Naval Institute Proceedings	Annapolis, Md.	xix
United States Steel Corp.	Steel	Pittsburgh, Pa.	i
W			
Wheeler, C. H.	Steam Condensers, Pumps, Deck Machinery,	19th & Lehigh, Philadelphia, Pa.	xviii
Y			
Yates-American	Heat Exchangers	Beloit, Wisc.	viii

How to cut cost and improve performance of heavy-duty equipment with USS "T-1" Steel

When you simplify fabrication . . . when you increase service life . . . when you cut down the need for repairs and maintenance, you save money. And these are precisely the ways in which you can save money with USS "T-1" Steel.

VERY HIGH STRENGTH

To the right, you'll find mechanical properties of USS "T-1" Steel that help you increase strength and durability without increasing weight . . . reduce weight and size without sacrificing performance.

TOUGH

The phenomenal toughness of USS "T-1" Steel—even at sub-zero temperatures—has been used to increase service life and minimize need for repairs in equipment that must withstand severe impact stresses outdoors in cold weather.

IMPACT AND ABRASION RESISTANCE

USS "T-1" Steel is now in service in coal chutes, ore cars, excavating equipment, and other jobs where abrasive wear and impact is severe and constant. It is doing an outstanding job: increasing service life and reducing repairs and maintenance.

HIGH-TEMPERATURE STRENGTH

This same USS "T-1" Steel serves in high stress applications at elevated temperatures. This creep-rupture data shows you why:

FABRICATION

USS "T-1" Steel usually doesn't require pre- or post-heating when welded or flame cut. With the proper electrodes, welds develop 100% joint efficiency without stress relief. This good weldability often permits faster, lower-cost field fabrication of heavy-duty parts.

USS "T-1" Steel can be furnished to the following heat-treated mechanical properties:

Plate Thickness:	¼" to 2" incl.	Over 2" to 4" incl.	Over 4" to 6" incl.
Yield Strength, Ext. under load (min)	90,000 psi	90,000 psi	90,000 psi
Tensile Strength (min)	105,000 psi	105,000 psi	105,000 psi
Elongation in 2", % (min)	18	17	16
Reduction of Area, % (min)	55*	50	45

**A standard .505" round tensile specimen is used if the thickness exceeds ¾". On sizes ¾" and under, a strap-type tensile specimen may be used which necessitates a lowering of the Reduction of Area specification to 45% minimum.*

All testing in accordance with ASTM recommended practices.

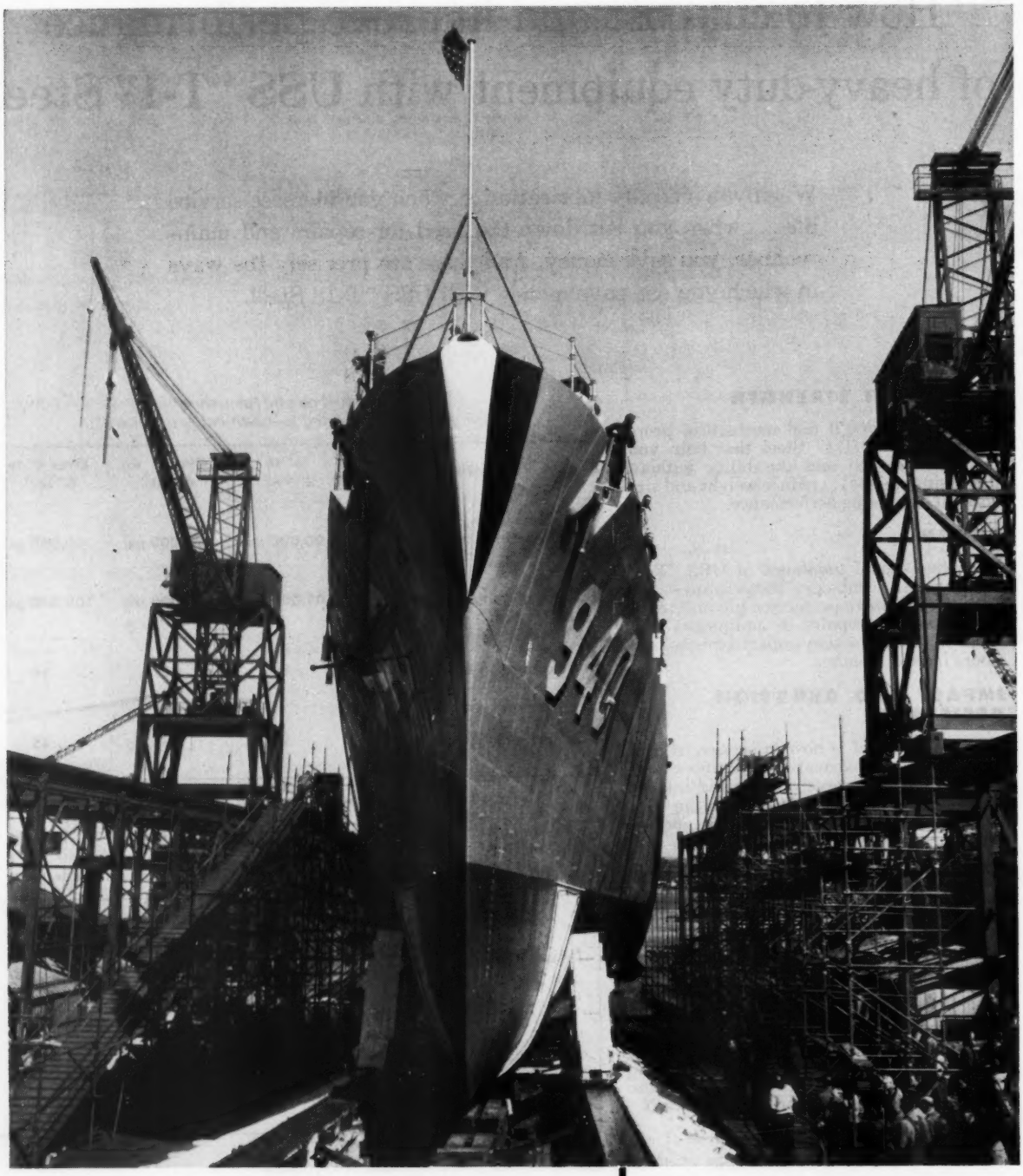
Test Temperature, °F	1% Creep in 10,000 Hr.	Stress 1,000 psi for 1,000 Hr.	Stress 1,000 psi for Rupture in 10,000 Hr.
900	44.0	59.5	44.0
1,000	10.5	23.0	13.8
1,100	2.3	10.0	5.8

USS "T-1" Steel can be drilled, machined, and cold-formed as easily as other steels of the same strength and hardness.

UNITED STATES STEEL CORPORATION, PITTSBURGH • COLUMBIA-GENEVA STEEL DIVISION, SAN FRANCISCO • TENNESSEE COAL & IRON DIVISION, FAIRFIELD, ALA.
UNITED STATES STEEL SUPPLY DIVISION, WAREHOUSE DISTRIBUTORS • UNITED STATES STEEL EXPORT COMPANY, NEW YORK

USS "T-1" CONSTRUCTIONAL ALLOY STEEL

UNITED STATES STEEL

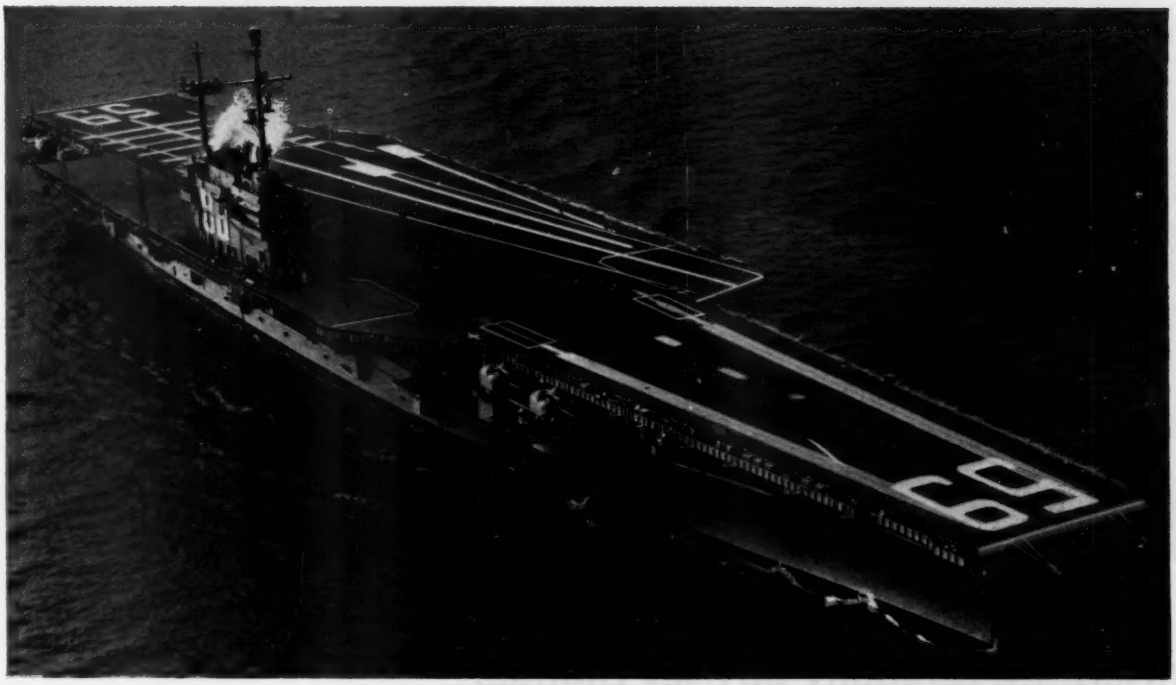


The U.S.S. Manley, DD940, sliding down the ways at Bath on April 12, 1956. The DD940, sponsored by Mrs. Arleigh Burke, is the second Bath-built ship named in honor of Commodore John Manley.

The first Manley, DD74, was launched at Bath August 23, 1917 and had a long and notable career in the active defense of our country in World Wars I and II.

BATH IRON WORKS
Shipbuilders & Engineers
BATH, MAINE

Where Equipment Must Be Dependable



U. S. S. Forrestal ★ U. S. S. Saratoga ★ Ranger* ★ Independence* ★ Kitty Hawk*

Allis-Chalmers equipment used by these carriers includes:

326 Centrifugal Pumps:

Fire Main Feed Booster
Main Condensate H.E.A.F. Blending
Mechanical Cooling Water Salt Water
Main Gasoline Catapult Water Brake

106 Drive Motors; Transformers, Main Turbine Condensers, Air Ejectors

*To be commissioned.

ONLY thoroughly proved equipment meeting the highest standards is given consideration for naval vessels. For super carrier service, where dependable performance and sturdy construction are *musts* — Allis-Chalmers supplies a broad line of major equipment.

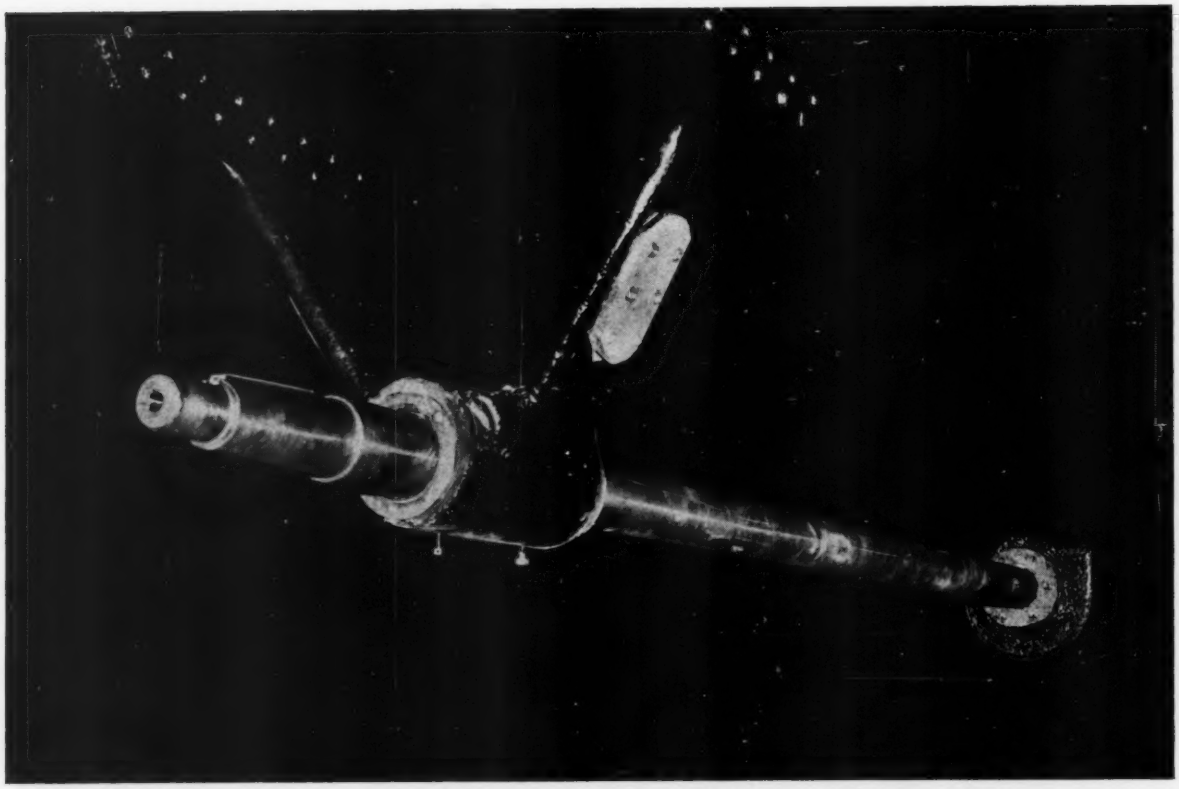
A-C has been supplying equipment engineered for Navy and Merchant Marine vessels for nearly 60 years.

For complete information about equipment for ships of all types, call your nearby A-C office, or write Allis-Chalmers, General Products Division, Milwaukee 1, Wisconsin.

Builders of steam turbine propulsion units, pumps, motors, control, transformers, condensers, air ejectors, deaerating heaters, diesel generators, turbine generator units, motor-generators, switchboards and control panels, metallic rectifiers and controllable pitch propellers.

A-5137

ALLIS-CHALMERS



Monel shaft and rubber bearings replace *Rio Escondido's* steel shafting (it wore) and babbitt metal bearings (they required hourly greasing). *Rio Escondido's* shafts and bearings were installed by Todd Shipyards, New Orleans, La.

Insurance company advises change:

Long-life Monel shafts replace steel on LST-turned-merchantman

Why fight a losing battle?

. . . Veteran seamen of Marina Mercante Nicaraguense asked themselves this question as they were planning the conversion of an LST into the merchantman *Rio Escondido.*

The ship — like all LST's — had bare steel tailshafts in Babbitt metal bearings. Pressure from grease lines kept sea water out. But Navy crewmen had to lubricate those lines. *Once every hour.* No easy task for a small crew in merchant service!

And the steel shafts wore in shallow water operation, despite this hourly battle against the sea.

It so happened that the ship's insurance company had experience with other converted LST's. They, too, had once had shafting troubles — but no longer. So the underwriter gave some good advice:

Solve your problem by replacing the steel shafting with Monel nickel-copper alloy shafts in rubber bearings. That way you eliminate greasing. You also avoid reboring the shaft log to take Monel sleeves on the steel shafts.

The owners followed this suggestion — and *Rio Escondido's* twin 5-in. dia. by 24½-ft. shafts — now of Monel alloy — don't cause a bit of trouble.

Of course they don't. Monel shafts give many owners *25 years and more* of trouble-free service. Extra strong and tough, they stay straight under tremendous shocks. Extra hard, they withstand wear, stay smooth longer. Extra stiff, too, Monel shafts resist whipping. And, in addition, there's the extraordinary corrosion resistance of this Seagoin' nickel-copper alloy.

* * *

No two ways about it! When it comes to long life and economy in metal equipment subject to rugged service and attack by marine corrosion, Monel is a Seagoin' metal. If *you* want help on any marine corrosion problem, write:

The International Nickel Company, Inc.
67 Wall Street New York 5, N. Y.

INCO NICKEL ALLOYS
TRADE MARK

Monel *"...it's the Seagoin' metal"*

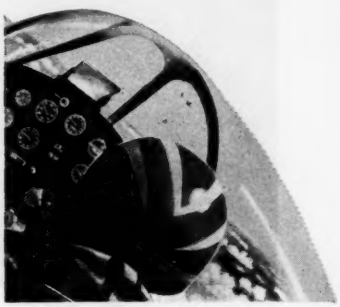
Monel and Seagoin' are registered trademarks of The International Nickel Company, Inc.

RCA's COMPACT, LIGHT-WEIGHT, HIGH-PERFORMANCE RADAR . . .

a long step toward the ultimate weapon

While the "enemy" plane is still far beyond the sight of human eyes, this new radar system detects it. Mounted in the intercepting aircraft, it provides a continuous flow of information about the "enemy position," electronically computed in terms of range and rate of closing. When visual contact has been established, the firing is controlled by the pilot, aided by electronic calculations.

In developing this compact, light-weight, high-performance radar, RCA has achieved a notable simplification of design with a minimum of components. As always, reliability and ease of maintenance have received top consideration. A feature is the bright radar display that permits viewing in broad daylight without a hood. Lock-on may be manual or automatic as the pilot desires.



OTHER MILITARY ELECTRONIC SYSTEMS DEVELOPED AND PRODUCED BY RCA INCLUDE . . .

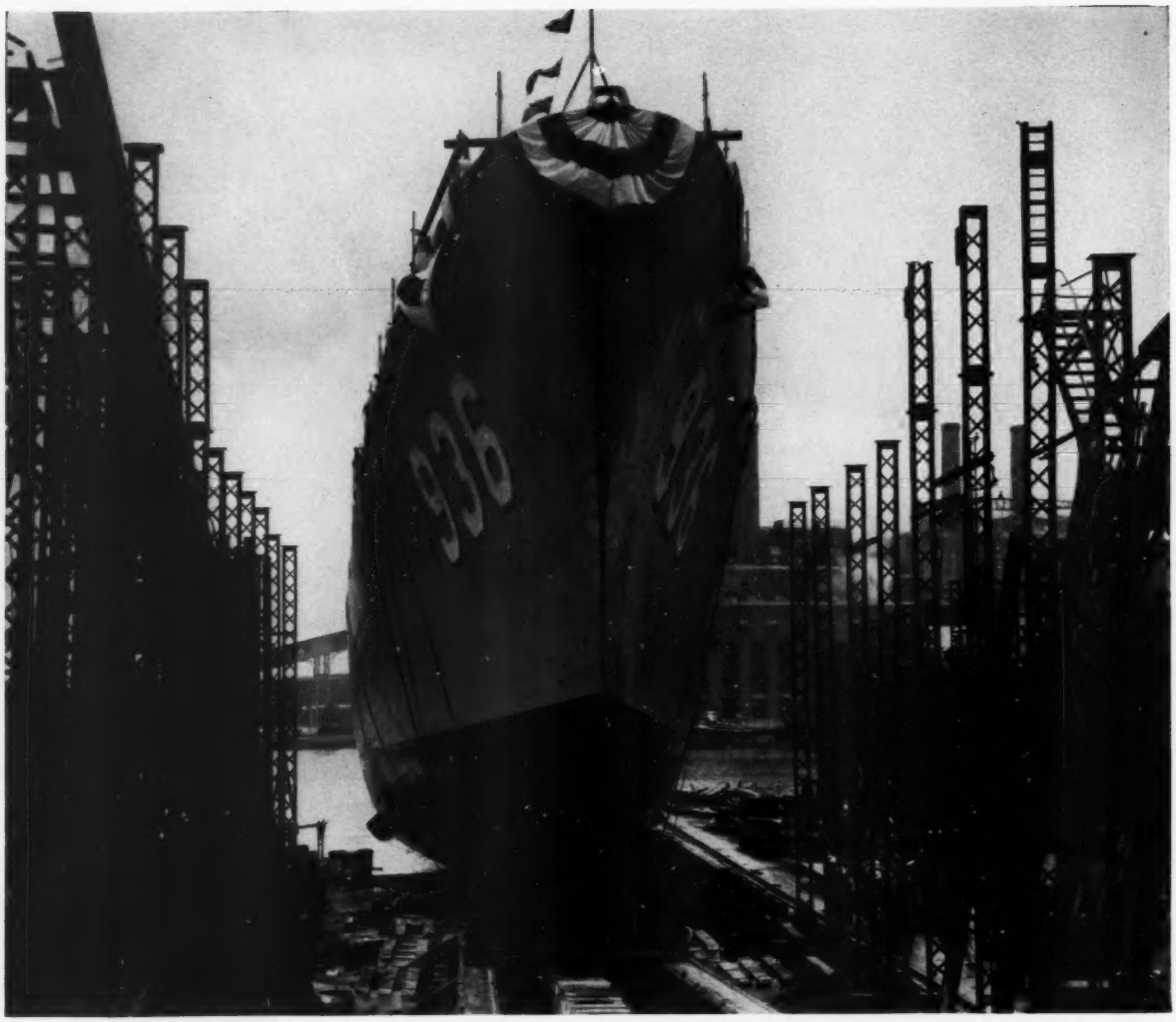
Radar: airborne, ground and shipboard

Communications: from hand-sized transceivers to 1.2 megawatt transmitters

Guided Missile Systems: including highly accurate tracking and instrumentation radar

Fire Control Systems
Loran
Bombing & Navigation Systems
Reconnaissance Systems
Infrared Equipment
Military Television
Computers

DEFENSE ELECTRONIC PRODUCTS
RADIO CORPORATION of AMERICA
CAMDEN, N. J.



The DD-936, named for Stephen Decatur, going down the ways at Bethlehem's Quincy Yard.

Names to Remember

STEPHEN DECATUR . . . scourge of the Barbary pirates . . . architect of the victory of the frigate *United States* over the *Macedonian* . . . a name that is part of the proud heritage of the U. S. Navy.

The builder of more than 800 ships for the Navy over the past 50 years, Bethlehem is part of that heritage and is grateful for the opportunity to serve the nation by constructing fighting ships with Names to Remember.

SHIP REPAIR YARDS

Boston Harbor — New York Harbor
Baltimore Harbor — Beaumont, Texas
Los Angeles Harbor — San Francisco Harbor

SHIPBUILDING YARDS

Quincy, Mass. — Staten Island, N. Y.
Sparrows Point, Md. — Beaumont, Texas
Terminal Island, Calif. — San Francisco, Calif.

BETHLEHEM STEEL

Shipbuilding Division

GENERAL OFFICES: 25 BROADWAY, NEW YORK 4, N. Y.

On the Pacific Coast shipbuilding and ship repairing are performed by the Shipbuilding Division of Bethlehem Pacific Coast Steel Corporation

Mechanical drawing class of Apprentice School students at Newport News. Those who excel advance to the technical drafting departments.

Shop training scene which shows an apprentice shaping a small die on a shaper, indicates how students are thoroughly trained.

How Skills are born...

at Newport News

Newport News utilizes the skills of almost every known trade...

And for the past three and a half decades, this Company's Apprentice School has furnished substantial numbers of the trained leaders and skilled craftsmen who make outstanding ships the tradition of Newport News.

This school's faculty... including experts for both shop practice and academic subjects... provides instruction equivalent to a Junior College education during four and five year courses.

Graduates numbering some 2200 skilled workers in 20 separate crafts comprise one more reason why quality identifies everything produced at Newport News.

Newport News

Shipbuilding and Dry Dock Company • *Newport News, Va.*

Condenser Tube Clinic

25 Years' Experience has proved the Value of Arsenic as an Inhibitor

Research at The American Brass Company nearly 25 years ago showed that about 0.04% arsenic would inhibit dezincification of brass, and this amount of the element has been added to Admiralty metal and aluminum brass since then.

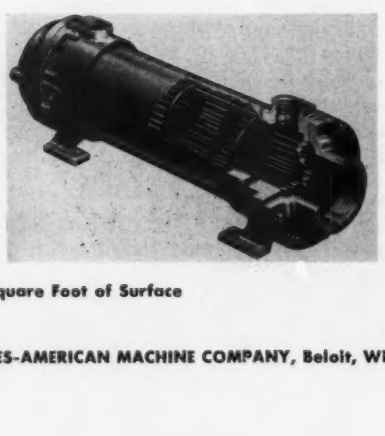
MARINE USERS generally find ANACONDA Arsenical Admiralty-439 satisfactory for inland-water vessels and also for seagoing vessels where service conditions are moderate; but ANACONDA Ambraloy-927 is preferred for most seagoing vessels. Both alloys are resistant to dezincification under all corrosive conditions found in these services.

YOUR OWN NEEDS may call for these corrosion-resistant Anaconda alloys or they may call for other tube alloys we make. Our Technical Department will be glad to help you select the one best alloy for the conditions you face.

TECHNICAL SERVICE—Get full information on our technical services and products by writing The American Brass Co., Waterbury 20, Conn. In Canada: Anaconda American Brass Ltd., New Toronto, Ontario. 5676B

ANACONDA® Tubes and Plates for Condensers and Heat Exchangers

Yates-American FIXED BUNDLE **HEAT EXCHANGERS**



for heating and cooling liquids

Proved by Test

GREATER CAPACITY Per Square Foot of Surface

Standard Sizes in 1, 2 and 4-Pass Designs

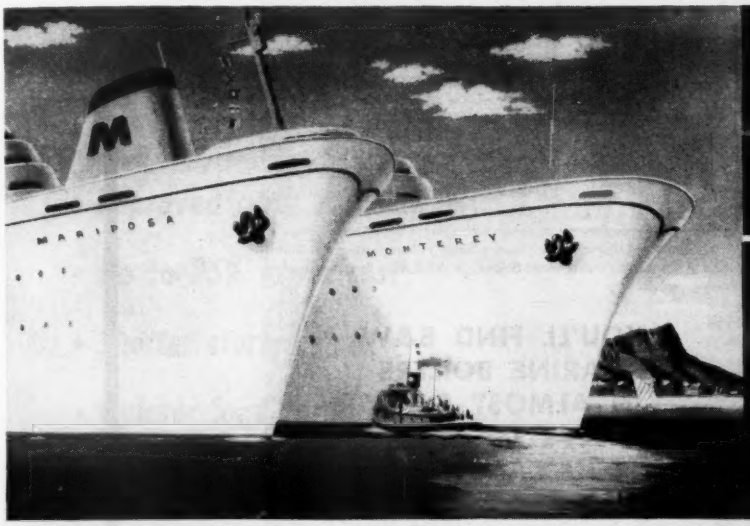
Also Manufactured to Customer Specifications

. . . All with exclusive feature of flanging baffles to eliminate vibration wear

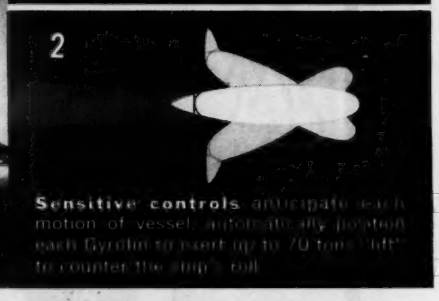
YATES-AMERICAN MACHINE COMPANY, Beloit, Wis.—Makers of Dependable Products since 1883

NEW MATSON LINERS COMPLETING INSTALLATION OF SPERRY GYROFINS

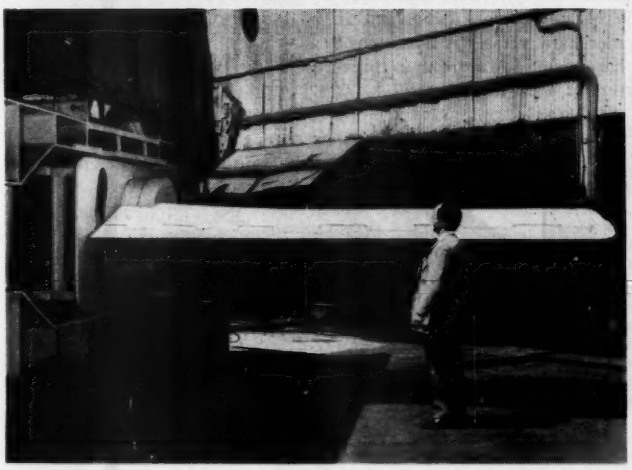
Monterey and Mariposa to sail this fall with new stabilizers



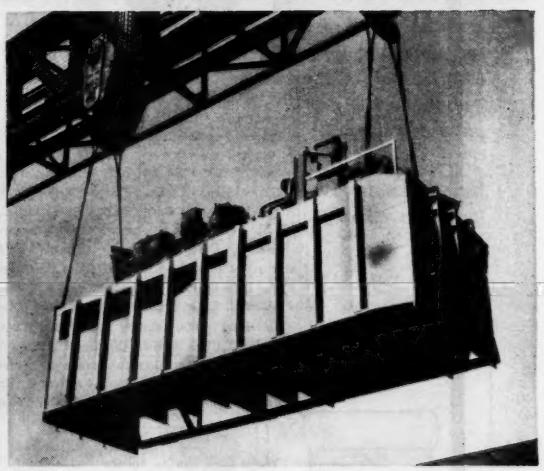
Compact design of Sperry Gyrofin* Stabilizers permits recessing into hull



Sensitive controls anticipate each motion of vessel, automatically position each Gyrofin to exert up to 70 tons "lift" to counter the ship's roll



Hydrofoil shape of fin, shown here before shipment to MARIPOSA, provides efficient lifting surface for stabilization.



Lowered into position, Gyrofin assemblies slip into openings in hull—one on each side, approximately midway between bow and stern.

In selecting Gyrofins for the Monterey and Mariposa, Matson becomes the first American line to offer its passengers a new comfort—relief from ships' roll regardless of rough seas.

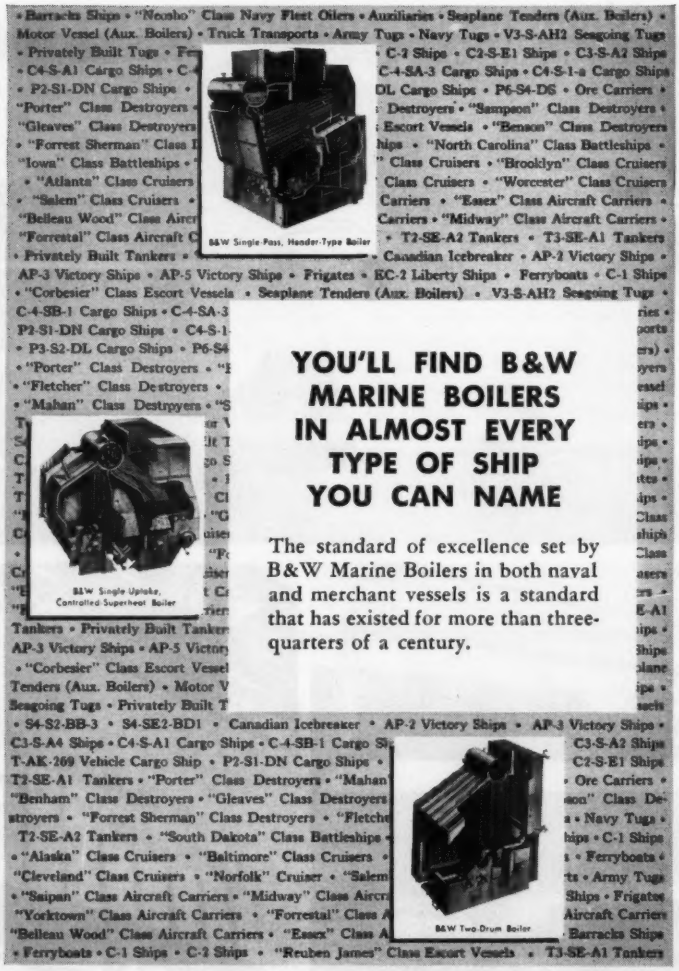
And in providing this new attraction to passengers, Matson will gain such other operating advantages as greater efficiency in maintaining schedules and minimizing strain on the ships themselves.

Sperry Gyrofins can be installed in your ships—new or old—to give you these same advantages. For information write our Marine Division or contact our nearest District Office.

*T. M. REG.

SPERRY GYROSCOPE COMPANY
Great Neck, New York
DIVISION OF SPERRY RAND CORPORATION

CLEVELAND • NEW ORLEANS • BROOKLYN
LOS ANGELES • SAN FRANCISCO • SEATTLE
IN CANADA: SPERRY GYROSCOPE COMPANY OF CANADA, LIMITED, MONTREAL, QUEBEC



YOU'LL FIND B&W MARINE BOILERS IN ALMOST EVERY TYPE OF SHIP YOU CAN NAME

The standard of excellence set by B&W Marine Boilers in both naval and merchant vessels is a standard that has existed for more than three-quarters of a century.

Water-Tube Marine Boilers
Superheaters . . . Refractories
Airheaters . . . Economizers
Oil burners
Carbon, Alloy and Stainless
Seamless and Welded
Tubing and Pipe
Welding Fittings and Flanges

BABCOCK & WILCOX

BOILER DIVISION

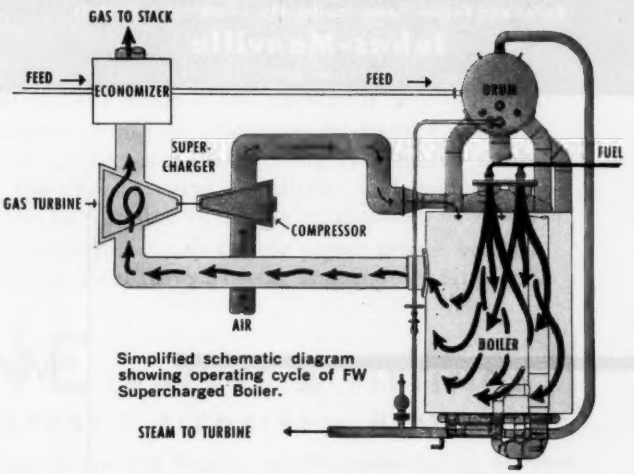
THE BABCOCK & WILCOX COMPANY, BOILER DIVISION
M-367 **161 East 42nd Street, New York 17, N. Y.**

New SUPERCHARGED Boiler

announced by Foster Wheeler

offers five important advantages for marine service

- *Up to 75% less weight*
- *Improved cycle economy*
- *Up to 25% lower cost*
- *Shorter startup time*
- *Quicker load response*



Simplified schematic diagram showing operating cycle of FW Supercharged Boiler.

FOSTER WHEELER has developed a new supercharged boiler of far reaching significance to the marine industry.

In the supercharged cycle, diagrammed above, high pressure combustion gases leaving the boiler are used to drive a gas turbine which, in turn, compresses combustion air for the boiler. Spent gases from the turbine pass through an economizer and thence to the stack.

The following advantages are inherent in this supercharged boiler cycle.

1. Due to higher heat absorption rates and a reduction in refractory of up to 95%, a supercharged boiler may weigh from ½ to ¼ as much as a conventional boiler of equivalent capacity.

2. Since combustion air is supplied by the supercharger, cycle economy is increased by the saving in power that would otherwise be required for operating the fans of a conventional boiler.

3. Smaller size and weight permits savings of up to 25% or more in first cost.

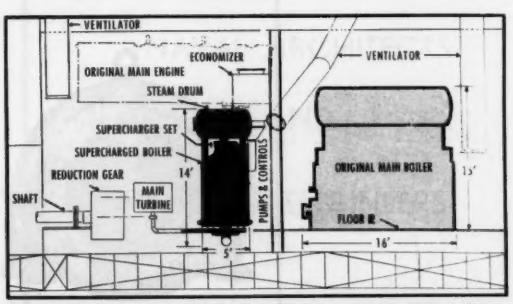
4. Small heat storage capacity results in extremely fast startup.

5. The supercharged boiler responds much more quickly to load changes.

The boiler unit is of exceptionally compact and symmetrical design, with boiler drum mounted above the furnace and connected to it by external risers and downcomers. The furnace is fully enclosed with waterwall surface and a metal baffle separates the furnace tubes from the convection section. Superheater pancake coils are located in the base of the firing chamber.

Foster Wheeler has built three supercharged marine boilers and a fourth is now under construction. All are for diesel fuel firing.

For complete information, write to *Foster Wheeler Corporation, 165 Broadway, New York 6, N. Y.*



Supercharged boiler (black) for driving a 6,000 SHP turbine in a Liberty Ship. Conventional boiler (gray) has been left in to provide a comparison of size and space relationship. Boiler and supercharger weigh 35,000 lb and occupy 60 sq. ft. of floor space. This compares with two conventional boilers weighing 144,000 lb each and occupying 156 sq. ft. of floor space.

FOSTER WHEELER

NEW YORK • LONDON • PARIS • ST. CATHARINES, ONT.

JOHNS-MANVILLE JM PRODUCTS

Materials for MARINE SERVICE

Incombustible Joiner Materials • Acoustical Materials
Asbestos Ebony for Switch and Panel Boards • Structural Insulations
Boiler and Engine Room Insulations • Packings • Gaskets

Johns-Manville
Box 290 New York 16, N. Y.

NAVAL ARCHITECTS
MARINE ENGINEERS

M. ROSENBLATT & SON

253 BROADWAY, NEW YORK
BEEKMAN 3-7430

The **TERRY TURBINE**

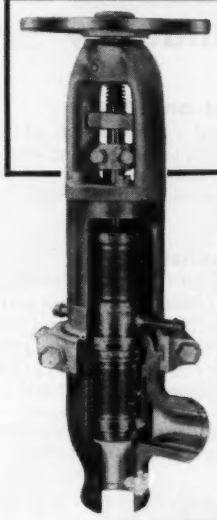
Terry Marine Turbines are dependable, compact, efficient machines especially designed for driving generators, boiler feed pumps, fuel oil pumps, compressors, etc. aboard ship. They offer the advantages of over fifty years experience in the Marine Field on both Commercial and Naval Vessels.

Turbines ranging from 5 to 3000 H.P. are built in the Terry Solid Wheel design as well as in the Axial Flow, single stage and multistage types.

Complete details on any turbine application will be gladly furnished.

THE TERRY STEAM TURBINE COMPANY
P. O. BOX 1200 **HARTFORD 1, CONNECTICUT**

T-1190



"OIL TO ATOMS"
brings this new look in valves

This is the type of valve found in the engine rooms of the *Nautilus* and the *Seawolf*, and in Navy's plans for added atomic-powered construction. The valve design and application are new, but the brand is well known for dependability.

From the start of nuclear power development, Crane was ready for the problems of controlling radioactive fluids. Crane had the skill in valve metallurgy and engineering. Crane had the background in quality production.

Today . . . tomorrow . . . as atomic power grows, you can count on Crane to meet Navy's needs. Your inquiry is invited.

CRANE VALVES & FITTINGS

PIPE • KITCHENS • PLUMBING • HEATING

Since 1855—Crane Co., General Offices: Chicago 5, Ill. Branches and Wholesalers Serving All Areas

PROVEN IN SERVICE

For 64 years, Cutler-Hammer, Pioneer Electrical Manufacturer, has furnished dependable control to all departments of the United States government. Built to specifications . . . backed by an outstanding record of performance.

CONTROL APPARATUS FOR ALL MARINE USES

Motor Control for Every Service, Ventilating Fans, Pumps, Cargo Winches, Capstans, Windlasses, Laundry Machines, etc.

Magnetic Brakes
Motor Operators for Valves
Limit Switches
Solenoids
Rheostats
Pressure Regulators
Magnetic Clutches
Pushbuttons

CUTLER-HAMMER, Inc., 1354 St. Paul Avenue, MILWAUKEE 1, WIS.

CUTLER-HAMMER
MOTOR CONTROL
C-H

GIBBS & COX, INC.

NAVAL ARCHITECTS

AND

MARINE ENGINEERS

NEW YORK

KAISER ALUMINUM products and engineering service for the marine industry

Sheet and Plate
Supplied as flat sheet, plate, coiled sheet, circles in a complete range of alloys, sizes and tempers. Specialty sheets available on request.

Extrusions
All standard alloys available in custom shapes (solid, semi-hollow and hollow), structural, rod and bar, and in extruded tube. Our two 8000 ton presses can produce extrusions up to 17 inches wide, 85 feet long, and up to 1,200 pounds in weight. In many cases, substantially wider extrusions can be produced, depending upon section design.

Forgings • Wire • Rod & Bar • Ingot • Tubing • Pipe • Foil • Electrical Conductor

Engineering Service—Our Marine Engineering Division will be glad to work closely with you to help solve your problems and develop improved designs. They will make available to you the extensive research and engineering facilities of Kaiser Aluminum. Conveniently located plants, including Trentwood, Wash.; Ravenswood, W. Va.; Halethorpe, Md.; Newark, Ohio. For immediate attention, contact one of our many sales offices. *General Sales Office,* Palmolive Bldg., Chicago 11, Illinois; *Executive Office,* Kaiser Bldg., Oakland 12, California.

Kaiser Aluminum

Call or write for our catalog "1956 PRODUCT DESCRIPTIONS AND AVAILABILITIES" and our brochure, "KAISER ALUMINUM HEAVY PRESS EXTRUSIONS."

See "THE KAISER ALUMINUM HOUR." Alternate Tuesdays, NBC Network. Consult your local TV listing.

Superstructure of this destroyer (Class DD931) is made of Kaiser Aluminum.

MILESTONES OF PROGRESS

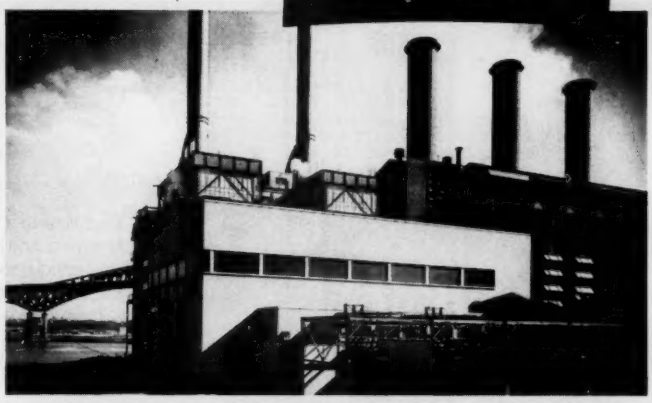
Highest Marine Steam Temperature — The SS Atlantic Seaman, one of three sister ships, is equipped with C-E Boilers designed to produce steam at 1020F — highest in Marine service.

. . . afloat

Highest Marine Steam Pressure — The SS Venore, one of a class of eight ocean going ore carriers, is equipped with C-E Boilers designed to operate at a steam pressure of 1450 psig — highest in Marine service.

Nuclear Powered Submarine — The USS Seawolf, one of the first of the Navy's nuclear powered submarines, will employ a reactor vessel and other related components supplied by Combustion Engineering.

and ashore



Highest Temperature, Highest Pressure — Ashore — Two C-E Controlled Circulation Boilers at the Kearny Station of Public Service Electric and Gas Company were the first utility boilers to be placed in service for steam conditions as high as 1100F at 2350 psig. Highest pressure and temperature projected to date will be 5000 psi and 1200F — a C-E Supercritical Boiler for Eddystone Plant of Philadelphia Electric Company.

• Combustion Engineering has long been identified with advanced standards of power practice — afloat and ashore. That's why you can expect top performance when you buy C-E Boilers.

COMBUSTION ENGINEERING

Combustion Engineering Building • 200 Madison Avenue, New York 16, N. Y.

Canada: Combustion Engineering-Superheater Ltd.

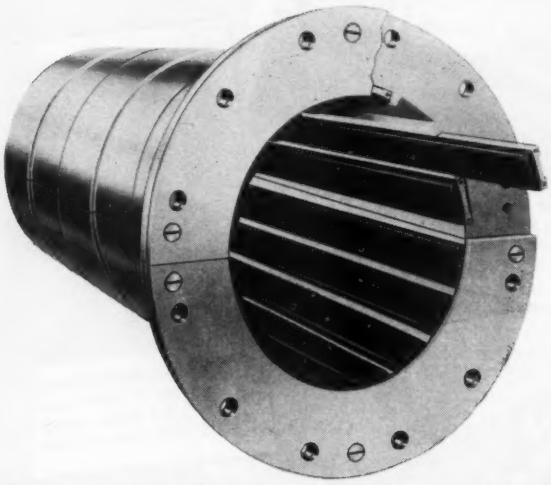
B-943

STEAM GENERATING UNITS • NUCLEAR REACTORS • PAPER MILL EQUIPMENT • PULVERIZERS • FLASH DRYING SYSTEMS • PRESSURE VESSELS • HOME HEATING AND COOLING UNITS • DOMESTIC WATER HEATERS • SOIL PIPE

CUTLESS BEARINGS

for

Stern Tubes and Struts



Soft rubber bearing surface—efficiently lubricated by water—this bearing far outlasts all hard surface types, protects propeller shafts, reduces vibration. More than pays for itself in extra wear alone. Saves you time, trouble, and upkeep expense.

LUCIAN Q. MOFFITT, INC.

Akron, Ohio

AIR-GROUND COMMUNICATION
POINT-TO-POINT COMMUNICATION
AIR NAVIGATION
AIRCRAFT INSTRUMENTATION
MISSILE GUIDANCE & CONTROL
COUNTERMEASURES
DATA TRANSMISSION
FIRE CONTROL
RADAR

Collins

CREATIVE LEADERSHIP IN MILITARY ELECTRONICS

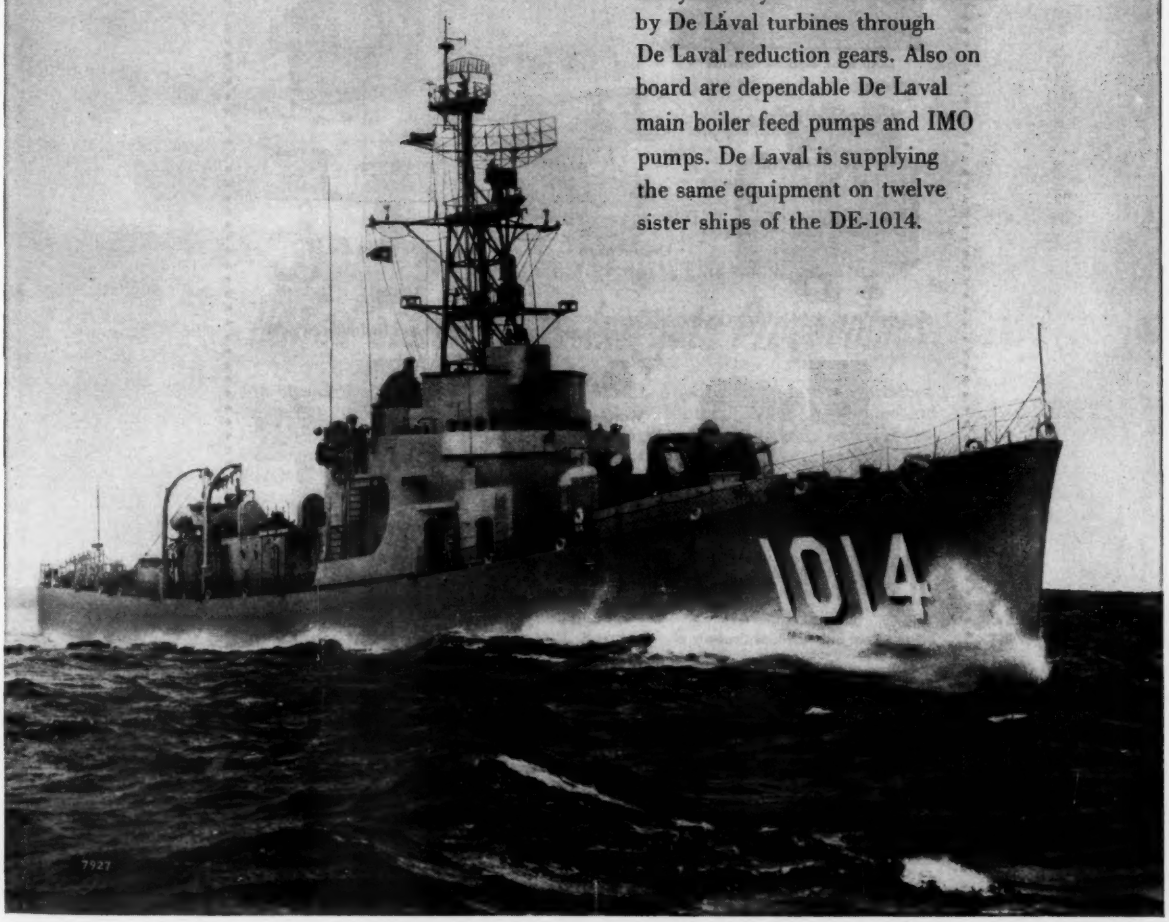
COLLINS

COLLINS RADIO COMPANY
Cedar Rapids • Dallas • Burbank
New York • Washington, D.C.

DE LAVAL

main propulsion units power the new United States Navy destroyer escorts

Here is the U.S.S. Cromwell, a Navy destroyer escort vessel driven by De Lával turbines through De Laval reduction gears. Also on board are dependable De Laval main boiler feed pumps and IMO pumps. De Laval is supplying the same equipment on twelve sister ships of the DE-1014.



DE LAVAL *Marine Division*

DE LAVAL STEAM TURBINE COMPANY

Trenton 2, New Jersey

DL 318

DE LAVAL

main propulsion units power the new United States Navy destroyer escorts

Here is the U.S.S. Cromwell, a Navy destroyer escort vessel driven by De Laval turbines through De Laval reduction gears. Also on board are dependable De Laval main boiler feed pumps and IMO pumps. De Laval is supplying the same equipment on twelve sister ships of the DE-1014.

DE LAVAL *Marine Division*

DE LAVAL STEAM TURBINE COMPANY

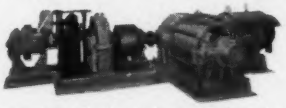
Trenton 2, New Jersey

DL 315

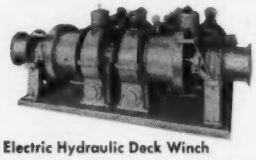
STEAM CONDENSERS...PUMPS
DECK MACHINERY...STEERING GEAR

and half a century of EXPERIENCE with all types of marine installations

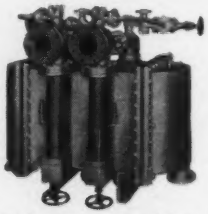
Dual Bank Steam Condenser



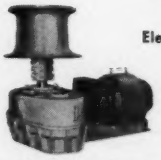
Automatic Tensioning Hoist — Airplane Crane



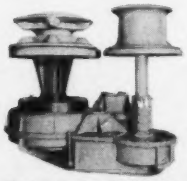
Electric Hydraulic Deck Winch



Tubejet Steam Jet Air Ejector

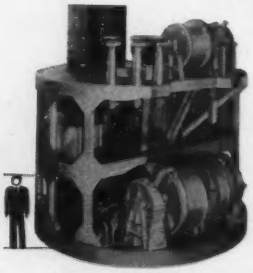


Electric Warping Capstan

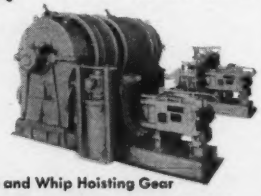


Electric Hydraulic Windlass

Warping Winch



Boat and Airplane Crane



Boat Crane and Whip Hoisting Gear

Steering Gears

WE.519

C.H. Wheeler
MANUFACTURING CO., 19th & Lehigh, Philadelphia 32, Pa.

Marine Condensers & Ejectors • Marine Pumps • Deck Machinery • Steam Condensers • Centrifugal, Axial & Mixed Flow Pumps • Steam Jet Ejectors • Vacuum Refrigeration • High Vacuum Process Equipment

Here is your authoritative source...

For information on the advances in professional, scientific and literary knowledge in the Navy and related services and professions . . .

Members of THE AMERICAN SOCIETY OF NAVAL ENGINEERS should read the

United States Naval Institute Proceedings

for pleasure and profit. The issues contain anecdotes and reminiscences, incidents from history and essays on Navy topics, technical articles and treatises on Naval developments and progress, book reviews and discussions, and international and professional notes. Membership dues (including PROCEEDINGS), $3.00 a year. Subscription rate, $5.00 a year. (Foreign postage, $1.00 extra.) Single copies 50 cents (except some scarce issues).

MAIL THIS MEMBERSHIP APPLICATION...TODAY!

U. S. NAVAL INSTITUTE - Annapolis, Maryland Date_____

I hereby apply for membership in the U. S. Naval Institute and enclose $3.00* in payment of dues for the first year, the PROCEEDINGS to begin with the_____issue.

I am a citizen of_____and understand that members are liable for dues until the date of receipt of their written resignations.

NAME: (Signature)_____

(Print)_____

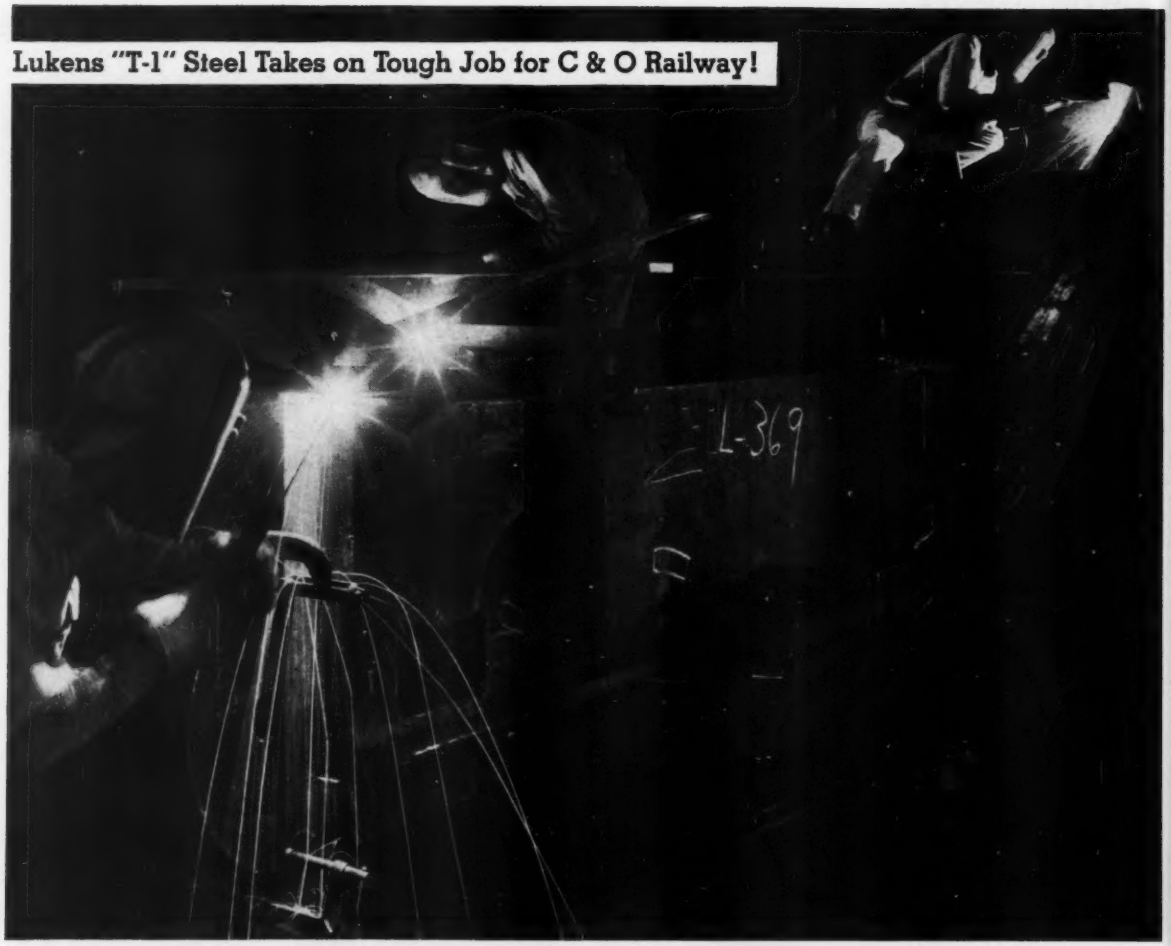
ADDRESS_____

_____

PROFESSION:_____

** $4.00 if residing outside of U. S., its possessions or territories.*

Lukens "T-1" Steel Takes on Tough Job for C & O Railway!



Horne Brothers of Newport News, first-line fabricators to the shipping industry, are building several new, improved coal trimming machines for the C & O Railway. Lukens "T-1" steel was specified to cut weight, combine strength and abrasion resistance. A number of other Horne Brothers' projects of Lukens "T-1" steel are already in the works.

Fabricator of unique new coal loader says:

"NEVER WORKED WITH AS TOUGH AND STRONG A STEEL THAT'S SO EASY TO WELD!"

■ *Enthusiastic* is the word for Horne Brothers' reports on Lukens "T-1" steel. They call it the best steel yet to combine strength and toughness, impact and abrasion resistance, and the all-important feature of excellent weldability.

Lukens "T-1" steel is economical, too. On this coal loading machine, for example, Horne Brothers was able to effect a 25% reduction in overall plate thickness—a direct result of this new steel's great strength. The finished machine will weigh far less than previous trimmers, operate with greater efficiency, less maintenance and increased life expectancy.

No wonder Horne Brothers is so well satisfied. No wonder they are planning to build more equipment of Lukens "T-1" steel—*right away!*

If your problem is equipment weight... abrasion, impact or atmospheric corrosion... the need for toughness at temperature extremes ... it will pay you to investigate this amazing new alloy steel now.

For the latest in fabrication techniques and specifications, write to Manager, Marketing Service, 821 Lukens Building, Lukens Steel Company, Coatesville, Pennsylvania.

LUKENS "T-1" STEEL

LUKENS

THE NEWEST IN A COMPLETE LINE OF ALLOY STEELS
LUKENS STEEL COMPANY, COATESVILLE, PENNSYLVANIA

Lakes' largest ore carrier, GEORGE M. HUMPHREY, is equipped with G-E propulsion and ship's service equipment.

"One source" buying from G.E. saves you detail, time, money

General Electric Marine Project Co-ordination eliminates costly, time-consuming, many-source purchasing; offers you complete planning, engineering, supplying and servicing of all propulsion and electrical auxiliary equipment for any marine installation. General Electric Company, Schenectady 5, N. Y. 200-106

Progress Is Our Most Important Product

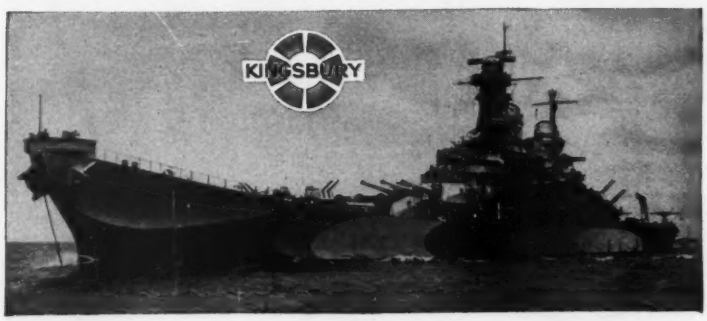
GENERAL ELECTRIC

KINGSBURY THRUST BEARINGS

Preferred through two World Wars for their dependability under difficult loads and speeds.

Each ship of "Iowa" Class has 36 Kingsbury Bearings, including the four main thrusts.

Kingsbury Machine Works, Inc. **Philadelphia 24, Pa.**



Aboard the USS Boston, too. . . .

COOPER-BESSEMER DIESELS

for emergency generation

● Converted into the U.S. Navy's first guided missile cruiser, the USS Boston (CAG-1), is one of many similar ships equipped with reliable Cooper-Bessemer emergency diesel generator sets.

Official U.S. Navy Photograph

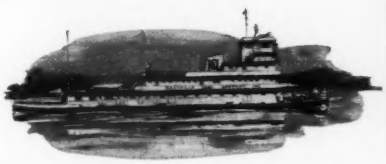
● Experience and advanced engineering has helped Cooper-Bessemer diesel engines provide an outstanding record of dependability aboard Navy and Coast Guard vessels ranging from the largest combat ships to relatively small patrol boats.
Find out for yourself the **new** things being done by one of America's **oldest** engine builders. Write for detailed information.

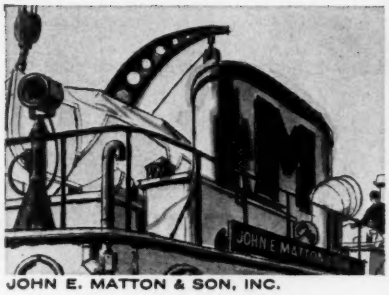
ENGINES: GAS • DIESEL • GAS-DIESEL • COMPRESSORS: RECIPROCATING, CENTRIFUGAL

cooper-bessemer

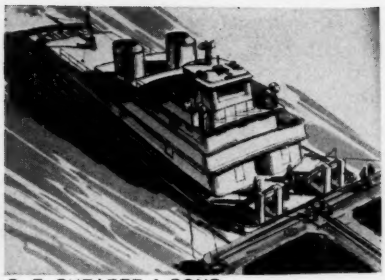
MOUNT VERNON, OHIO • GROVE CITY, PENNA.

NASHVILLE COAL CO., INC.





JOHN E. MATTON & SON, INC.

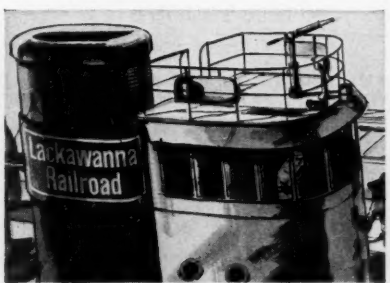


O. F. SHEARER & SONS

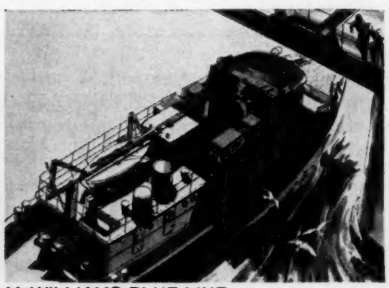
THE POWER: GM TURBOCHARGED DIESEL

AND THE STORY— Since January, nine leading towboat and tugboat operators have specified General Motors Turbocharged Diesels for their new boats. Because a GM Turbocharged Diesel gives them 75% more power, with virtually no increase in engine size and a big reduction in weight per horsepower. And, horsepower for horsepower, a GM Turbocharged Diesel costs less to install, run and maintain. Get full details on the GM Turbocharged Diesel for tankers, tugs, cargo ships, towboats, dredges, or any other application, from your local Cleveland Diesel Engine Division representative, or write direct.

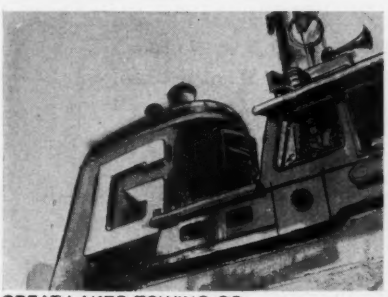
GREAT LAKES DREDGE & DOCK CO.



D., L. & W. RAILROAD



McWILLIAMS BLUE LINE



GREAT LAKES TOWING CO.

CHOTIN TOWING CO.

BROWN & ROOT INC.

CLEVELAND DIESEL

An Engine Division of General Motors · Cleveland 11, Ohio

SALES AND SERVICE OFFICES:

Boston, Mass., 9 Commercial Ave., Cambridge, Mass. Tel.: Eliot 4-7891
Chicago, Ill., 216 West Potomac Ave., Lombard, Ill. Tel.: Randolph 6-9214
Dallas, Texas, 9404 Waterview Road Tel.: Fairdale 2403
Honolulu, T. H., 3115 Diamond Head Road. Tel.: Honolulu 99-9202
Miami, Fla., 2315 N. W. 14th Street Tel.: 64-2852
New Orleans, La., 727 Baronne St. Tel.: Magnolia 6761
New York, N. Y., 10 East 40th Street Tel.: Murray Hill 5-4372
Norfolk, Va., 554 Front Street Tel.: Madison 2-7147
Orange, Texas, 212 First Street Tel.: Orange 8-4226
Pittsburgh, Pa., 469 Marlin Drive Tel.: Locust 1-2173
Portland, Ore., 3676 S. E. Martins St. Tel.: Prospect 1-7509
St. Louis, Mo., 2 N. Wharf St. Tel.: Main 1-0642
San Diego, Calif., 3886 Sequoia Street Tel.: Hutson 8-6883
San Francisco, Calif., 870 Harrison St. Tel.: Douglas 2-1931
Seattle, Wash., 1230 Westlake Ave. N. Tel.: Adler 1440
Wilmington, Calif., 433 Marine Avenue. Tel.: Terminal 4-4098

GM CLEVELAND DIESEL

Copyright 1956, by the American Society of Naval Engineers, Inc.

Journal of the AMERICAN SOCIETY of NAVAL ENGINEERS, Inc.

VOLUME 68 AUGUST 1956 NUMBER 3

Statements contained in articles herein are the private opinions and assertions of the writers and should therefore not be construed as reflecting the views of the AMERICAN SOCIETY OF NAVAL ENGINEERS, INC., nor of any other organization with which such writers are affiliated. The Society as a body is not responsible for statements made by individual members.

COUNCIL OF THE SOCIETY

(Under whose supervision this number is published)

Rear Admiral KENNETH K. COWART, USCG, President

Rear Admiral RAWSON BENNETT, USN
Captain PETER V. COLMAR, USCG
Rear Admiral R. E. CRONIN, USN
Rear Admiral L. V. HONSINGER, USN
Captain PETER HORN, USN
Commander E. G. KINTNER, USNR
Mr. JAMES S. MELTON
Lieutenant Commander J. W. SAWYER, USNR

Mr. THOMAS H. SHEPARD, JR.

Captain J. E. HAMILTON, USN (Ret.), Secretary-Treasurer

SECRETARY'S NOTES

Council Meeting

Progressively, our Council appears to be required to handle more important Society business with each successive meeting. It is difficult to explain this on the basis of Society growth since this, though positive, is neither rapid nor accelerating. Perhaps a generally complicating social and economic situation is responsible and maybe the reason is unimportant to the members of the Society. Regardless of the reason it is a fact. Various matters which were considered at the Council meeting on 29 May 1956 are discussed below.

Banquet Considerations

For the second time, the Society has two possible locations for its annual banquet. From 1946 through 1955 the banquet was held at the Statler Hotel. This year the new banquet hall of the Sheraton Park Hotel was tried out. The problem which faced the Council was to decide between the two facilities for 1957.

The first action, which was taken without dissent was to decide that there would be a banquet in 1957. The two facilities were considered on these criteria:

a. The dining room
b. The dinner itself
c. The many receptions given before and after the banquet
d. The location of the hotel

These matters were considered by the 1956 Banquet Committee, which reported its opinions to the Council, but made no decision. The Council and the

Banquet Committee both *unanimously* agreed that:

a. The large dining room at the Sheraton Park was superior to any room or combination of rooms at the Statler, *but* that we had crowded too many people into it this year. Sheraton Hall, for serving 1700 to 1800 people is the best location in Washington. (This year we served 1955)

b. Although individual experience and opinion vary a great deal, it was agreed that there can be no valid choice between the two hotels on the basis of quality of food or service. They are essentially equal.

c. The second floor layout, plus the monolithic construction of the Statler provides a greatly superior arrangement for the receptions (or hospitality rooms) than the horizontal dispersion of the Sheraton Park does.

d. The downtown location of the Statler is somewhat superior for local traffic (parking, etc.) and for convenience of hotel reservations at other hotels.

With no disagreement on the only criteria which the Council believed had any bearing on the problem, the Council evaluated these and made its decision that the 1957 banquet would be held at the Statler Hotel.

We believe that this was an important decision as far as the membership is concerned and for that reason this full report is being made.

The decision was based almost entirely on the third criterion. The essential feature of the ASNE banquet, which has always made it especially enjoyable, has been the pleasure of renewing acquaintances. Visiting many of the hospitality suites before and after dinner, encountering former associates, shipmates and friends, and meeting new people, is the distinctive motif of the party. The Council feels that its decision best preserves this.

1957 Banquet

As stated, the 1957 banquet will be held at the Statler Hotel in Washington. The date is Friday, 3 May 1957. Formal notices in regard to reservations will be mailed in the normal way in January or February 1957.

One other detail which the Council decided is of considerable importance to many members. *This is that reservations will be accepted on the basis of at least* TWO MEMBERS for EACH NON-MEMBER guest. This decision is a compromise between the former one-for-one and the proposal that we limit attendance to members only. The time has come when something must be done to check the growth in reservations. It is believed that the two-for-one may accomplish this without the necessity of rejecting any otherwise valid applications.

As was done in 1955, when the total number of reservations reached 1907, all usable space in the Statler will be utilized for the dinner. This will again bring up the complaint of many that they were not seated in the main room. Based on its best judgement, the Council believes that it is better to accept the maximum number of reservations than to turn away several hundred who cannot be located in the main room.

We will know a great deal more about the wisdom of these decisions on 4 May 1957.

The Journal

The Council decided to continue the Journal in the same format and approximately the same size as the first two 1956 issues. The only change directed was a modification in the coloring of the Society seal on the cover. The Council feels very elated that the "new" Journal has elicited a great deal of favorable comment with only one criticism: that it could not be folded and comfortably carried in a back pocket. We had expected to receive six or eight comments when the format was changed, since that is a considerably larger number than any previous measure has brought forth. The receipt of about forty written and many verbal comments, all favorable (except that one) is a very gratifying and satisfying thing.

The Society Seal

The Society Seal was designed when the Society was founded, we believe. However, we can find no evidence of any official color design for the seal. The decision to use color on the Journal cover brought this issue to a head. The coloring on the cover of this issue is believed to be the proper one. It is expected that the Council will select the coloring, which will become official, from the two cover (May and August) designs.

Progress of Growth and Maturity

In the February 1956 Secretary's Notes we carried, under this head, a description of the changes in administrative matters. This brought us the following very welcome letter from the Society's Oldest Living Past Secretary-Treasurer, Captain Henry C. Dinger, U.S.N., Retired:

"24 W. 55 St.
New York 19, N. Y.
June 7, 1956

SecTreas, ASNE;

I am submitting the enclosed comment regarding the article in the Feb. 1955 issue of the Journal. If it is considered too long or some of the remarks too unsympathetic suitable editing or deletions are acceptable but I think if possible the actual facts of the day of not so long ago should be presented.

I also believe some of the comments later than 1910 can be corroborated by your experience. I just opened May number and am enclosing yellow sheets. Congratulations on contents of last two numbers. May comment.

Best Wishes
Henry C. Dinger
Capt., U.S.N., Ret.
Oldest Living Ex Sec-Treas."

Captain Henry C. Dinger, U.S.N., at the time that he was Director of the Engineering Experiment Station.

The comment first mentioned will be found under "Discussion" on page 609. The yellow sheets mentioned are Captain Dinger's additions to the Secretary's Notes on "Progress of Growth and Maturity." These are printed below. In a few cases we have added comments in the form of footnotes which are indicated by numbers in parenthesis:

Captain Dinger's Memo relative to
Society History. Received June 12, 1956
"JOURNAL, ASNE Secretary's Notes

Banking, Pg. 211. I was in Bureau and worked with Edwards on Society matters. The Bank was American Security and Trust Co. where account was kept for many years.[1]

Society's Office, Pg. 212. The work was done by the Secretary-Treasurer, who received 75 dollars a month. Some typing help or copying was occasionally done by Bureau Clerks, usually by the clerk assigned to Bureau officer who was Secretary-Treasurer. At home, valiant assistance was given by the various wives.

I was a sort of volunteer second or third assistant during the terms of Edwards, Reed, White and Mallory. The Journal storeroom was in the sub-basement and partly in a basement corridor of the old State, War and Navy Building. The desk,[2] which I trust is still in existence, was moved from home to home. I acquired it from Fenton before end of his term about September 1908, and turned it over to Holmes. Holmes bought a card index case[3] and put the accounts from the books on cards. This was about the time Fessenden came into the picture. He had been hired during Fenton's regime, but I believe I was the first Secretary-Treasurer he worked with.[4] The Council meetings were held in the small room to the north of the main drafting room. This was Edward's office in 1903, when he was compiling and editing the Fuel Oil Report, and was the office of Dinger, Holmes and Church while Secretary-Treasurer. The various Presidents from 1903-1918 were not all[5] stationed in the Bureau, and usually came to Secretary's office for meetings, conference, etc. Here the first Society banquet was organized in 1909, in the days or soon after the Babylonian (Newberg) Captivity when Admiral Capps was chief of both Bureaus. The original idea of having a banquet was launched at a lunch attended by William D. Hoxie of B. and W. Co., (host), Struther Smith (member of Council) and myself, and probably also C. W. McAllister, Chief Engineer, Revenue Cutter Service (now USCG). I am not absolutely sure about the latter, but he also was a member of Council banquet committee, and did valiant pioneer work, particularly in lining up connection with civilian engineers, the press, and governmental authorities who might be interested in Naval Matters. So there was a sort of political engineering in the concept of these banquets which have grown like the mustard seed to such size that apparently difficulty is now experienced in finding a space to house them.

Pg. 213, second desk. A slight error, the card index case procured by Commander Holmes (Fessenden will bear witness). There was also a chair.

I note comment regarding clearance to publish, and would say that the fact that for 35 years the Journal did publish engineering information from the universe to the universe without propaganda, but on scientific and factual basis has given it a very enviable status, something like, "the casting of bread upon the waters."

[1]We have learned that a Society checking account was opened in the American Security and Trust Co. (our present bank) in 1891. This bank opened for business in 1890. We are still in ignorance as to how the Society's funds were handled during the first two or three years.

[2]This desk was sold to a clerk named Charles S. Woodin, apparently on April 2, 1913. He paid five dollars on that date and three dollars on May 16, 1913. Record is not too clear due to destruction of "Journals", one of which shows a credit to "Furniture" on this date.

[3]The procurement of card case is correct. However, it happened in 1910, which is clearly shown by entries on the earlier cards. Incidentally, the transferring of names to cards was at Mr. Fessenden's suggestion. They were transferred from mailing lists in the form of books.

[4]Mr. Fessenden agrees that this is correct.

[5]Mr. Fessenden believes: "I think this in error, though my time in Bureau commenced on January 25, 1906. All Presidents listed 1903 to 1918, except John D. Ford, were known to me in Bureau of Steam Engineering."

This appears to me to be startlingly confirmed by the book review of Harold Bowen's book in May 1956 number. I could, however, cover some of the queries the reviewer makes on matters pre-Bowen, and incidentally, Harold Bowen did his first tour of engineering duty under me on the USS *Maryland*, 1905. After I peruse Bowen's book I may attempt this.

I regret to hear of Miss Beresford's passing. She was one of the main helps of the Journal, even prior to 1907. I am also happy to notice Lanman Engraving are still with you.

Use any of this as you may deem appropriate.

(Signed) H. C. Dinger"

Journal Engraving

Captain Dinger's final comment brings in a sad note. At the 29 May meeting, the Council directed that the engraving for the Journal be shifted to the Gulbenk Engraving Co. in Nashville, Tennessee. This was *in absolutely no way* a reflection of any kind on the Lanman Engraving Co. which, in the present Secretary-Treasurer's experience, is a superb company, with an excellent product, which always leaned over backward to help the Society in every way. The change was based on convenience and economy in matters over which Lanman had no control.

The severance of old ties is a most difficult thing and after the many years of such happy and satisfactory relationships, the Secretary-Treasurer's task of telling Lanman Engraving Co. of the Council's decision was a most unpleasant one.

Mr. John C. Niedermair

Since Mr. Niedermair was a member of the Society's Council in 1951 and 1952 we were very much elated by the announcement that he was among the ten outstanding public servants who had been selected to receive a Career Service Award. The Bureau of Ships Daily Administrative Bulletin for 27 June 1956 reported this in the following words:

"Mr. John C. Niedermair, Technical Director of Preliminary Ship Design, Bureau of Ships, has been selected by the National Civil Service League to receive one of its Career Service Awards for outstanding public service, the first Navy candidate to be so recognized. The Navy's chief naval architect since 1938, Mr. Niedermair has guided the basic designs from which some 8,000 combatant and auxiliary vessels have been constructed. He has been called the father of today's modern Navy, for almost all the proud ships that won World War II at sea as well as the mighty units joining the fleet today bear the stamp of his genius.

The Niedermair award is one of ten granted nationally each year by the National Civil Service League as a part of its efforts to enhance the prestige of public service.

Born in Union Hill, New Jersey, in 1893, Mr. Niedermair attended grade and high schools in Staten Island, New York. He won a scholarship to the Webb

MR. JOHN C. NIEDERMAIR

Institute of Naval Architecture and Marine Engineering from which he graduated in 1918. That same year he began his remarkable career as a ship draftsman in the New York Navy Yard.

Mr. Niedermair's stature permeates the entire field of naval architecture. His work on watertight integrity and ship stability has left a lasting imprint on practically every merchant ship constructed since 1929. He served as a consultant to a Senate Subcommittee investigating the *Mohawk* and *Morro Castle* disasters. In the course of the construction of the liner *America*, Mr. Niedermair was appointed by Admiral Emory S. Land as technical conciliator between the design agent and building yard.

Despite his impressive contributions to commercial naval architecture, Mr. Niedermair's first love and the field of his greatest triumphs has been combatant ship design. Besides the direction and inspiration his leadership has provided BUSHIPS naval architects, Mr. Niedermair has originated many critical design innovations. His conception of the high tensile steel flight deck structure combined with his idea of building ballistic decks in layers to compensate for the then existing limitations in welding techniques, made possible some of the most valuable military characteristics of the *Essex* class aircraft carriers. Again, on the afternoon of November 4, 1941, in response to a British requirement that day, Mr. Niedermair made a small pencil sketch and calculated the basic characteristics of an entirely new and radical ship which became the workhorse of World War II. Less than a year later on November

2, 1942, LST-384, first of a class which eventually totalled more than 1,100 ships, was commissioned, a feat of ship design and construction rarely if ever equalled.

Perhaps Mr. Niedermair's greatest contribution to his country will be realized through his efforts on behalf of his profession and through the young engineers he has trained and developed. He has been very active in professional circles and particularly in the Society of Naval Architects and Marine Engineers. He has fostered a high degree of technical excellence and professional pride in the men he has worked with and has done much by precept and example to inspire them and instill excellence into their work. He has wisely counselled two generations of naval officers. Almost every high-ranking officer of the Navy now engaged in ship design work has learned the fundamentals from "Johnny." "

For his exceptional achievements during World War II, The Civilian Distinguished Service Medal was awarded to Mr. Niedermair by Secretary of the Navy Forrestal.

In 1951 while a member of the Council, Mr. Niedermair represented this Society and the Society of Naval Architects and Marine Engineers at the International Conference of Naval Architects and Marine Engineers in London, England. He presented a paper entitled "Ship Motions" at that conference.

Captain Peter Horn, USN

Captain Horn, a present member of the Council, has been assigned to command the Naval Research Laboratory, Bellevue, D. C. With Admiral Bennett on the Council the Society can now boast of having on its directorate the Navy's two principal military researchers.

Official U. S. Navy photograph

Captain S. M. Tucker, USN, (left) congratulates Captain P. H. Horn, USN, the new Director of the U. S. Naval Research Laboratory in Washington, D. C. Captain Tucker, a native of Jackson, Mississippi, is scheduled to retire shortly, completing 30 years' service. Captain Horn, of Philadelphia, Pennsylvania, received his Master of Science degree from the University of California. He is a member of the American Society of Naval Engineers, Society of Naval Architects and Marine Engineers and of the Armed Forces Communications and Electronics Association.

Brand Award—1956

The Admiral Charles E. Brand award is given to the officer student at MIT standing first in Course XVII for the year. The 1956 recipient is Lieutenant Thomas J. Walters, USN.

Mr. Thomas H. Shepard, Jr.

The Editor wishes to apologize publicly to our Council member, Mr. Thomas H. Shepard, Jr. In the first two issues of the Journal following Mr. Shepard's election, we have carried his middle initial as "B". This is one of those gremlin-controlled things where efforts to correct just did not work out. We will open this August issue with trepidation until we are assured that the "B" has become an "H".

Honorary Membership—Mr. K. D. Williams

At the annual banquet a certificate of Honorary Membership in the Society was awarded to Mr. K. D. Williams. However, for reasons of health, Mr. Williams deemed it inadvisable to attend. As a consequence, the Secretary-Treasurer made the actual presentation at Mr. William's home.

The Secretary-Treasurer of the Society presents a certificate of Honorary Membership in the American Society of Naval Engineers to Mr. K. D. Williams.

American Society of Naval Engineers Awards

There is pending before the Council a proposal that an annual award be made by the Society to the "most promising engineer" in the graduating class of certain of the service academies. We are corresponding with the authorities of these academies in an exploratory way. The matter will be considered by the Council at a later meeting.

Junior Membership

A too literal interpretation of the by-laws provision in regard to Junior Memberships has caused a minor administrative problem. To remove this the Council has agreed to the following interpretation:

Any member of the senior classes at any of the designated schools is eligible for junior membership from the day he successfully completes his Junior year until his graduation. Any Junior Member who is accepted prior to 1 January of his Senior year will have his membership dated on the date of acceptance. Any who are accepted after 1 January will have his membership dated as of 1 January. Junior Memberships will lapse on the second 31 December following graduation. Junior Members will receive copies ot the Journal from date of membership. In all cases the period of Junior Membership as determined above will be covered by the payment of two annual dues at one half the regular rate.

Thus it is possible for a Junior Member, who takes advantage of his earliest eligibility, to actually be a member for two and one half years, and to receive copies of ten issues of the Journal at a total cost to him of one year's regular dues. We believe this to be an exceptional opportunity which members would do well to call to the attention of junior eligibles.

Advertising

At the present time, the Society is drawing on its reserve funds in order to produce the improved Journal. The Council wishes to continue, even to improve, the Journal, but it is loath to suggest to the membership that the annual dues be increased. Only once in its 68 years have the Society's dues been raised and the present rate is only fifty percent higher than the original 1888 rate. During that interval all costs have increased considerably more.

We feel that a Journal has now been developed which, with its circulation of more than 5,000 could be an attractive advertising medium for hundreds of industrial companies. Our rates are probably lower than could be justified. A moderate increase in advertising income will permit the Society to continue to operate at its present level without increasing the dues.

To test this, under Council authorization, we have engaged the David Industrial Art and Ad Agency, Inc., of New York City as medium representative. The arrangement is set as a test for one year (four issues). It is hoped that this will result in regular income sufficient to eliminate the draft on the reserves and to permanently table a move to increase the dues.

Errata

We know that a number of errors, mostly-proofreading, appeared in the May Secretary's Notes. We have no excuse but will try to do better.

Back-Issue Journals

We have never made a business of furnishing copies of back issues of the Journal. Occasional demands have been satisfied if possible. Normally we do not maintain much of a stock except of current issues. Recently we acquired a complete set of Journals from 1889 through 1952. These are bound in annual volumes and are in good-to-excellent condition. This set will be sold to anyone who wants it subject to prior sale. Any offer will be considered, but the Society reserves the right to refuse any offer which is considered to be too low.

Courtesy of New York Shipbuilding Corp.

U. S. S. JOHN WILLIS (DE-1027)

Ready for launching at New York Shipbuilding Co., Camden, N. J. on 4 February, 1956.

Navy Ship Building Program

A Navy Press release on 23 July 1956 announced the 1957 shipbuilding and conversion program which was recently approved by the President following enactment by Congress. Included in the release was information on the awards to builders of some of the ships. Others plus some remnants of the 1956 program are yet to be awarded.

The program and the present status of awards is:

Attack Aircraft Carrier (Forrestal-Class) (CVA 64)
1 to New York Naval Shipyard

Nuclear-powered guided missile cruiser (CLGN)
1 to Bethlehem Steel Co. to be built at Quincy, Mass.

Nuclear-powered submarines (SSN)
1 to Portsmouth Naval Shipyard
1 to Mare Island Naval Shipyard
4 to be awarded on a competitive basis

Guided missile Frigates
2 to Philadelphia Naval Shipyard
2 to be awarded on a competitive basis

Guided missile destroyers (DDG)
8 to be awarded on a competitive basis

Escort Vessels (DE)
2 to be awarded on a competitive basis

Ammunition Ship (AE)
3 to be awarded on a competitive basis
(2 of these are carry over from the 1956 program)

The following Service and Landing Craft are to be awarded on a competitive basis:

5 Patrol Vessels (YP)
13 Landing Craft Utility (LCU) (10 of these in 1956 program)
2 Large Harbor Tugs (YTB)
950 Landing Craft Vehicle, Personnel (LCVP) (850 of these in the 1956 program)
126 Landing Craft Mechanical (LCM(6) (64 of these in the 1956 program)
64 Landing Craft Mechanized (LCM(3) (all in the 1956 program)
350 Landing Craft Personnel, Large (LCPL) (240 of these in the 1956 program)

1 Landing Craft Mechanized (LCM) Hydrofoil has been awarded to the Norfolk Naval Shipyard for construction.

The Ship Conversion Program is just as impressive as the new construction program. This Conversion Program is made up of:

1 Attack Transport (APA) to be awarded on a competitive basis

Guided Missile Light Cruiser (CLG)
1 to Bethlehem Steel Co., Quincy, Mass. Plant
1 to Bethlehem Pacific Coast Steel Corporation
1 to New York Shipbuilding Corporation
1 to New York Naval Shipyard
1 to Boston Naval Shipyard

Attack Aircraft Carrier (CVA)
1 to Newport News Shipbuilding and Drydock Co.
1 to New York Naval Shipyard
1 to Puget Sound Naval Shipyard
1 to San Francisco Naval Shipyard

Amphibious Assault Ship (LPH)
1 to New York Shipbuilding Corporation

Seaplane Tender (AV)
1 to Philadelphia Naval Shipyard

Radar Picket Escort Vessels (DER)
2 to Philadelphia Naval Shipyard
2 to Boston Naval Shipyard
2 to Long Beach Naval Shipyard

Ocean Radar Station Ship (YAGR)
2 to Philadelphia Naval Shipyard
2 to Charleston Naval Shipyard

Miscellaneous Auxiliary Vessel (EAG)
1 to Norfolk Naval Shipyard

The above programs are included in the revolution whose management Admiral Mumma discusses on page 425.

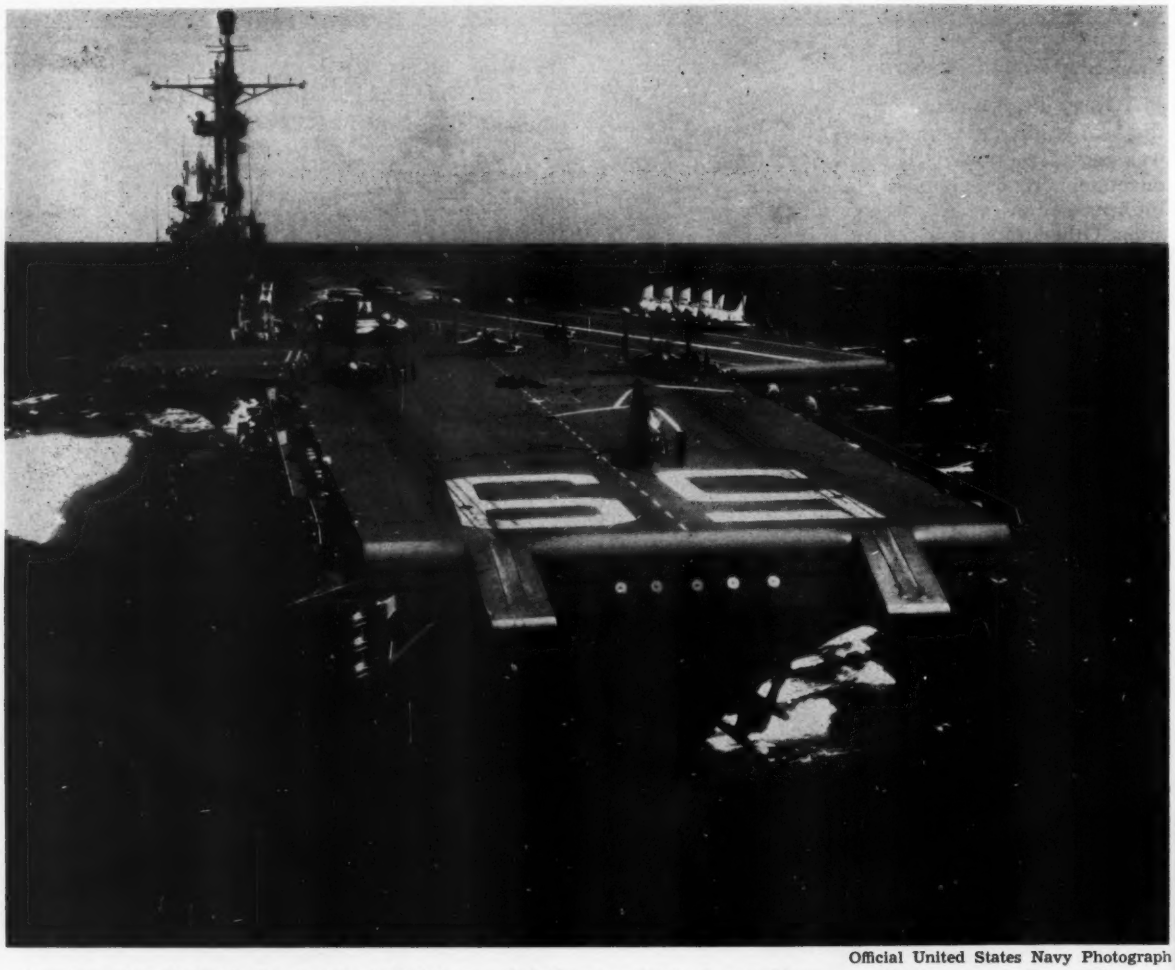
Symposium on Naval Hydrodynamics

A *Symposium on Naval Hydrodynamics,* sponsored by the Office of Naval Research (Mechanics Branch) in cooperation with the National Academy of Science-National Research Council, will be held in Washington, D.C., from September 24 through 28. This Symposium—the first of a series to be held on a yearly basis—is an outgrowth of a continually growing need for meetings devoted exclusively to developments in the rapidly advancing field of hydrodynamics, particularly in the areas basic to Naval and marine applications. Emphasis will be on critical surveys of these various areas, including identification of requirements for further research, to be presented by recognized authorities. An important aspect of these meetings is the stimulation of interest in research in classical hydrodynamics by making known more widely the many scientifically fruitful and technologically important problems extant. This first meeting will be unclassified and open to all those interested.

The program for the first Symposium includes the following speakers: C. C. LIN on boundary layer stability, S. CORRSIN on turbulence in shear flows, G. K. BATCHELOR on wave scattering due to turbulence, M. J. LIGHTHILL on river waves, W. H. MUNK and M. TUCKER on the ocean wave spectrum, D. GILBARG on free streamline theory and steady state cavitation, M. S. PLESSET on physical effects in cavitation and boiling, H. SNAY on hydrodynamics of underwater explosions, G. P. WEINBLUM on seaworthiness, H. LERBS on hydrodynamics of marine propulsion, J. V. WEHAUSEN on ship wave phenomena, M. STRASBERG and H. M. FITZPATRICK on hydrodynamic noise, J. B. PARKINSON on hydrodynamics of water-based aircraft, J. W. MACCOLL and R. N. COX on basic hydroballistic phenomena, and J. C. NIEDERMAIR on hydrodynamic barriers in ship design.

The program will include tours of U .S. Navy laboratories in the Washington area.

Information on registration and other arrangements may be obtained by writing to Mr. John S. Coleman, National Academy of Sciences-National Research Council, Washington 25, D.C.



Official United States Navy Photograph

Low bow view of the USS Forrestal (CVA-59) on a shakedown cruise in the Caribbean Sea off Guantanamo Bay, Cuba.

Official United States Navy Photograph

The USS John Paul Jones (DD-932), during sea trials, second ship of the Forrest Sherman class, steams at high speed near Penobscot Bay off the coast of Maine.

REAR ADMIRAL A. G. MUMMA, U. S. Navy
Chief of the Bureau of Ships

MANAGING A REVOLUTION

The following article is reprinted from the May 1956 issue of Armed Forces Management. The author as Chief of the Bureau of Ships of the Navy Department is responsibly as close as any other individual, to the extreme technical advances which accelerated research and development have made possible to the armed services.

Evolution equivalent to a revolution is taking place in the Navy. The engineers' dream of atomic power has become a reality in the *Nautilus* (SSN-571). Now, only 1 year after her trials, the Navy has in its current and proposed building programs an atomic fleet of 14 other submarines, a cruiser, and designs and preliminary component work on an aircraft carrier.

Far-ranging missiles are just as rapidly supplanting guns. Two submarines and five cruisers already have guided missile capabilities, and current and proposed programs provide for the construction or conversion of a whole fleet of guided missile ships.

These two developments, as historic as the introduction of steam and the rifled gun, are being integrated with other developments which, taken collectively, are also revolutionary.

For instance, the extraordinarily successful teardrop shape of the *Albacore* (SS-569), designed for high speeds and great maneuverability underwater, will be used for the latest seven of the nuclear powered submarines. The nuclear-powered aircraft carrier will utilize brand new, extremely powerful catapults and the advanced angled-deck principle.

The Bureau of Ships and the field activities under its management control play a key part in this revolution. We are the Navy's technical agent for shipwork. We design, test, procure, and maintain ships' hulls, machinery, and equipment ranging from washing machines to radar. We are responsible for research and development associated with these primary tasks and in various collateral fields such as salvage and radiological, chemical, and biological warfare defense.

We also procure certain vessels for the Army and Air Force, and act as procurement agent for ships to be transferred to friendly foreign nations. As Bureau Chief, I am responsible for the Department of Defense and, in wartime or periods of declared national emergency, act as Coordinator of Conversions and Repair for that Department and represent it in coordinating work with other Government agencies. The Bureau is responsible for mobilization planning in these various connections.

It takes a big outfit to do this job. The Bureau's headquarters in Washington is manned by about 4,000 people. Outside of Washington are 11 naval shipyards, several laboratories and other technical facilities, and a variety of other activities spread all over the world. Altogether our organization consists of over 100 separate activities employing over 125,000 civilians and 5,000 military personnel. We comprise one of the 10 largest businesses in the Nation.

BIG BUSINESS

This is big business, but it is not civilian business. Ours is a military support organization, as responsive to military command and discipline as the operating forces. The key positions are manned by specially trained officers.

These men are given extensive technical and management training which equips them extremely well to manage an industrial organization. The majority of them are engineering duty specialists who have postgraduate degrees in engineering or naval architecture. They are given a series of carefully selected assignments carrying progressively more responsibility and having great variety.

Throughout their careers their practical work is augmented by lectures; courses, and seminars on various aspects of management. Some of our officers are given high level management training at postgraduate schools of such institutions as Harvard, the University of Pennsylvania, the University of California, and others. Thus, to each new assignment, they bring not only ever broadening experiences, but also increasingly various contacts and fresh points of view.

By the time they are occupying positions of substantial responsibility, they have acquired an extensive knowledge, both practical and theoretical, of management techniques.

Actually, officers fill only a relatively few key management and staff positions. Civilians usually provide the highly specialized skills and man the intermediate levels. We spend a lot of time, effort, and money assuring appropriate training for these people.

This is a big job. There are, for instance, over 2,000 separate occupations in the shipyard alone. Training in these occupations, which range from welder through engineer to managerial official, is the responsibility, first, of the immediate supervisor. When the requisite training is beyond his capacity, he calls upon the very extensive resources of the activity and occasionally upon facilities available outside the activity.

These resources and facilities deal with a great variety of matters. For instance, recently many persons have been given special courses in the controlled maintenance program, a management tool that will be discussed later. Others have been trained in industrial fund accounting methods. Many have participated in our 4-year apprentice program or our 5-year engineer development program. A few have been selected to attend postgraduate schools in management at various universities.

A great many of our supervisors, some of whom are engineers more used to dealing with verifiable facts than unpredictable people, have participated in a program designed to help them deal effectively with human beings. An indication of the extent of our efforts is that 4,000 different courses were given in 1 year at the shipyards alone; there were about 160,000 participants in these courses.

A continuous program for training on such a scale is required because procedures and techniques are constantly changing, particularly in a revolutionary era. Through the program we are assured that our people have the skills and knowledge to do their work and do it well.

These are the people who run the Bureau and its field activities. One of our concerns is to see that they work in logical organizational units. Fortunately, this is not a difficult matter insofar as the broad outlines are concerned. That part of management which must be centralized in Washington is the Bureau's task; production work is either supervised or done by the field activities.

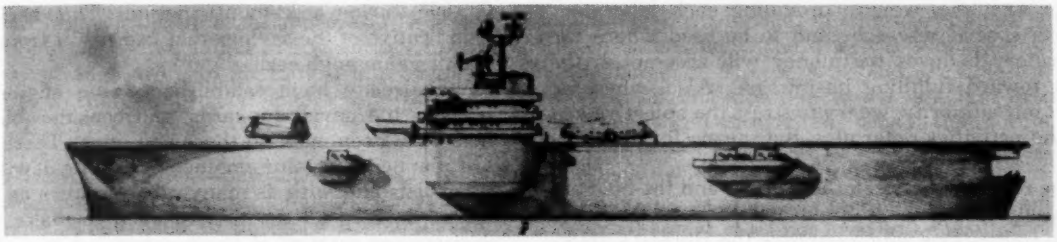
Generally, it is not difficult to define clearly the responsibilities of the field activities, because the tasks for which they are established usually form logical and rather simple organizational units. Thus, we have a shipbuilding scheduling activity, a mine defense laboratory, a boiler and turbine laboratory, and a radiological defense laboratory, among others. Their missions are well indicated by their names.

We have supervisors of shipbuilding to represent the Bureau at private shipbuilding yards, industrial managers to represent us at ship repair yards, and a few inspectors of machinery to represent us at the plants of some machinery producers. (Generally we are represented at manufacturers' plants by the material inspection service of the Office of Naval Material.)

The naval shipyards are our most complex field organizations, for their tasks are varied and difficult to compartmentalize, especially without detriment to efficiency or national security. They are large, usually the largest industrial organizations in their respective localities. Their tasks range from routine supply functions to the construction, conversion, overhaul, or repair of a great variety of naval ships which are packed with complicated electronics and ordnance gear.

SPECIALTIES

Almost every yard has some specialty. The Portsmouth Naval Shipyard, for instance, specializes in submarine work. At each shipyard trades are organized into shops, such as the electrical shop, enabling the shipyard to have shops appropriate in size, number, and type to its work.



Navy helicopter assault carrier (LHP) as conceived by artist is included in the proposed 1957 conversion program and is designed for landing assault troops by helicopter.

In charge of each of these shops is a shop master. The shop masters report to a production officer, who directs and coordinates their work. This breakdown is designed for jobshop work, which comprises the overwhelming portion of shipyard work. However, when a repetitive job comes along, we can and do set up an *ad hoc* assembly line.

Over-all management of all these activities and their work is one of the jobs of the Bureau in Washington. In addition, we have the people and machinery to decide what work should be done and where. This includes research and development planning, ship design, assigning work to field activities or contracting for its accomplishment by private firms, control of the development of components, such as radar, and myriad other tasks.

Here the organizational problem is most difficult, for the work requires special technical skills and close functional coordination, both to a great degree, and the two are not always compatible.

EIGHT DIVISIONS

Our current answer to the organizational problem is a breakdown into eight divisions, four of which are devoted to the technical functions of the Bureau: research, development, and design; shipbuilding and maintenance; electronics; and nuclear propulsion.

One division handles the Bureau's responsibilities in connection with the field activities.

The remainder are service divisions which handle such matters as finance, contracting and internal services, personnel matters, and management. The head of each of these divisions is delegated great authority, leaving the Chief of the Bureau relatively free to handle matters transcending any one division.

One key division of the Bureau is the shipbuilding and maintenance division, which is broken down into four subdivisions. One subdivision is made up of ship-type branches such as the submarine branch, carrier branch, auxiliaries branch. which are responsible for the ship as a whole. Two subdivisions, the hull and the machinery subdivisions, are concerned with technical features common to many or all ships, such as hull arrangement, boilers, and turbines and gears.

The remaining subdivision handles such functions as inventory control, logistics, cataloging. This breakdown appears to afford the Bureau the greatest opportunity for exploitation of specialized skills while providing the functional coordination so essential.

The actions of all the Bureau's divisions affect the field activities in one degree or another. However, the managerial aspects of the Bureau's relations with the field are usually handled by or channeled through the field activities division.

One of the more important aspects handled by this division is the essential controls that must be maintained by the Bureau. A primary control is inherent in our assigning work and money. Another is a limited control over personnel matters, particularly over the numbers of persons at each activity.

The Bureau also requires sufficient uniformity in organization, methods, procedures, personnel practices, and the like, to provide a general framework for effecting management improvements in all activities, and to provide a similarity of operations within the activities dealing with the Fleet that will make it easier for the Fleet and the various Navy Bureaus and offices to deal with them.

In addition, we require comparability of accounting methods. This enables us to present intelligible financial reports to higher authority. It also affords a good tool for comparing cost of performance. Financial control is further exerted by keeping close tabs on expenditures as a check on estimates for a particular task or subdivision thereof, thus affording good bases for re-estimates such as are necessary.

Control of quality and quantity of work is afforded through reports and reviews from a variety of sources, varying from the reports of commanding officers of ships undergoing overhaul to reports required from the field.

All these controls are useful to the Bureau in making decisions regarding budget formulation, allocation of funds, redistribution of personnel, the recommending of military billet distribution, and the directing of the shipment of certain materials.

We do our best to keep these controls down to the essential minimum. The people commanding our field activities are capable, and they know best the local situation and the best way to accomplish their activity's mission. The more freedom they have, the better they will do their job. We do our best to extend this area of freedom.

For example, we are providing each laboratory with a special research fund to be used where the laboratory believes the money will accomplish the most toward fulfilling its mission. Also, when we exert our authority, we try to stress the objective and let the activities determine the means for reaching it.

The relations between the Bureau and the field are definitely "two way." Field activities continuously make suggestions which improve the Bureau's methods of doing business. Frequently one activity develops a good management tool which the Bureau then disseminates to other activities. When we learn of a management technique from, for example, private industry, we study its applicability to our activities. When the idea is applicable, we tailor it as necessary and disseminate it to the field, from which we in turn learn much about how it can be improved.

This interplay between the Bureau and the field activities has resulted in the development of a large number of excellent management tools in recent years.

For instance, we now have a controlled maintenance program for public works departments at naval shipyards. This program stresses the concept of preventive maintenance. It involves planning and estimating by a centralized staff—for all work except emergencies and service calls—plus continuous inspection and the scheduling of work so as to utilize labor fully, avoid material delays, determine work backlog, and provide a measure of performance. It is also designed to eliminate excessive duplication and over-maintenance.

This program has been operating for some time at the Charleston Naval Shipyard. The overhead cost has been very small, and already the program is paying large dividends. Savings are made in production time and costs by the virtual elimination of breakdowns. Savings in the costs of major overhauls are being made through the adequate inspection and maintenance the program provides. We are now proceeding to install the program at all the naval shipyards.

Another management technique rather recently developed is what we call value engineering. For some time the Bureau has been concerned with the rising cost of naval ships. Recently, the Bureau became aware of the advantages of organizing a separate staff section to apply the test of value to design, manufacture, and purchasing techniques, and to educate other sections in this technique.

Value engineering may be defined as the study of a necessary function so that it can be accomplished at minimum cost and without impairing required performance and maintenance standards. It involves a critical review of existing satisfactory ways of doing things so that equally satisfactory but less costly alternatives can be developed and exploited.

During 1 year key areas of the Bureau of Ships actively participated in the application of these principles and developed ideas that will save about $18,000,000 during a 12-month period. This beginning is indicative of the savings that can result from wide use of value engineering.

We already have established value engineering units at the naval shipyards, and considerable savings have been made in those activities. These units are manned by value engineers who deal with the specialists and with as many other sources as possible to develop all the facts needed to make sound decisions on matters of engineering and cost. When the value engineer comes up with a recommendation, it is presented to the responsible specialists; but its feasibility must be proved to the satisfaction of those specialists before it is adopted.

Management is far more than special tools and techniques or organization and methods, of course. I believe that the key to good management is to instill in all hands a basic progressive attitude. We want thinking and planning in terms of the future, as well as the present.

This is particularly important during a revolutionary period, when even in doing one's day-to-day work a man can get the erroneous impression that he is being forward-looking, when the normal pressure of day-to-day work seems to leave little time for thinking and planning, when complacency is likely to set in, and when the rapidity of development makes planning for the future all the more important.

Certainly the rapid movement of current events demands careful planning for the future. The extensive and rapid introduction of nuclear power and guided missiles aboard our naval ships is bound to have complex and widespread impacts upon those aspects of the Navy's business with which the Bureau of Ships is concerned. The sooner we can discern these impacts and take appropriate steps, the better.

We have found that the only effective way to do this is to take the time to think and plan, to consider the problems at top level meetings, and to see that the people at those meetings consider such problems with their subordinates. The farther down we go in soliciting thinking and planning, the more detailed and complete the results become. By the time the ideas filter back up, a sound approach has been evolved and can then be implemented.

But this does not happen of its own accord; it requires pushing by top management to prevent its being lost in the immediacy of day-to-day pressures.

PROGRESSIVE ATTITUDE

We want our people to have a progressive attitude toward their technical work as well. To obtain it, we have had to take steps to reduce conservatism. Too often in the past we have been dubious of taking chances. We have insisted on waiting until we have all the facts, carefully proven, before putting a new development into production.

Too often we have concentrated almost exclusively on a short-term program for achieving moderate

progress because the program appears to be a sure thing. We have often failed to concentrate upon a more long-range program for radical improvements which is, in the light of good engineering judgment, a sound risk because we want to commit ourselves only on a sure thing.

New ideas and radical approaches to problems are available, particularly in the minds of our younger personnel. They need to be given encouragement and solid backing before they will bring these ideas into the open and push them. They need to be told by their superiors that their ideas are welcome. Their superiors must be willing to take the necessary risk, to make a commitment.

I do not believe that this approach is so daring as it might at first appear. A technical agency such as ours has a certain quota of "washouts" in its progressive programs. All too often these failures have occurred even when nothing significantly new or daring was attempted. The percentage of failures will not increase significantly as long as good engineering judgment is used. Certainly far greater progress will be made.

A high degree of receptiveness to new ideas is bound to engender some highly improbable suggestions. These might be considered a waste of time by some. However, I think that such ideas should be given serious consideration by a variety of people, not only because there might be some application for the idea that no one had thought of before, but also because mere exposure will further facilitate receptiveness to radical departures.

For instance, the idea of a surface ship that steams along with its hull below the water and its superstructure jutting into the air is not new, but few have given it any serious consideration for naval application. We have given serious consideration to this apparent oddity and discover that it may well have an extremely valuable application.

We also urge that new ideas be given positive as well as serious consideration, despite immediately apparent and considerable objections. Recently, for example, it was urged that the Bureau change two conventional submarines already in the advance stages of construction into guided missile submarines.

Such a change is not to be taken lightly. It involves considerable additional expense and delay in completion, disruption of orderly building procedures, the sacrifice of certain military characteristics in favor of missile handling capabilities, and other factors. In the past such formidable objections might well have been considered adequate justification for a negative response to the idea.

However, in line with our approach, the "objections" were regarded as problems to be overcome, not insuperable obstacles, and careful and specific estimates of what was involved if the change were made. As a result, apparently insuperable obstacles were revealed to be in fact no more than problems which could be measured and weighed against the advantages of having missile capabilities aboard these submarines. The final decision was to make the change.

With all this attention to the new and dramatic, one might well ask what happens to the routine, day-to-day, plodding efforts? It seems to me that there need be few, if any, such efforts. A problem dealing with, for example, the malfunctioning of one type of naval engine does not need to be routine. It will not be routine if it is recognized as an opportunity to develop new engineering knowledge.

We are encouraging our engineers and contractors to go beyond merely solving such a problem. We are urging them to study it from all points of view, to surround it and extract every iota of new engineering knowledge, to use the knowledge obtained as a starting point for future progress.

The "routineness" of some projects is due to their apparent unimportance. We are taking a look at such projects to determine whether or not they really are important and contributing to progress. As soon as engineering judgment indicates that they are not, we drop them, even though $1,000,000 more might complete a $5,000,000 program. When a project is determined as profitable, we note our error in having considered it unimportant and push it as hard as we can. "Routineness" and a progressive attitude are incompatable.

This progressive attitude cannot be "institutionalized" in a busy organization. It takes emphasis on the part of top management, constant willingness to give serious consideration to new ideas, and willingness to make a commitment, to take a risk based on sound engineering judgment. Now that a significant portion of our people are thinking in terms of the future, this progressive attitude is a great deal less difficult to maintain. Nevertheless, this precious asset must be guarded with vigilance, lest it succumb to routine and immediate pressures.

Capable, well-trained people thinking and operating in terms of the future, in a sound organization with sound management procedures and techniques —this is our answer to the problem of managing a revolution.

It will be noted that this method makes no direct mention of controlling a revolution, of preventing it from getting out of hand.

There are two basic reasons for this lack of emphasis. One is that we are a military organization in the midst of a technological race for superiority. It is difficult to proceed too rapidly in this circumstance. The other is that sound engineering judgment, which we never permit to be sacrificed in the interest of our progressive attitude, prevents forays into profitless areas.

The most dramatic technical revolution in a century is upon us. It did not overwhelm us. The Bureau's organization and procedures were sound enough to absorb the first shock, and to permit the necessary adjustments without disruption or delay.

Nor are we simply riding the crest of the wave.

Encouraged by the new progressive spirit of our management, our people are more than ever before pushing new installations, ferreting out new applications and implications, and imaginatively creating new ideas. They are enthusiastic and their enthusiasm is accelerating the revolution in naval ships, extending it, intensifying it.

This is the way we are managing this revolution. This is how we are creating today the ships of tomorrow.

* * * * * * * * * * * * * *

U. S. NAVY

SUBMARINE SHIPBUILDING PROGRAM

Hull	*Yr.*	*Type*	*Name*	*Builder*
571	52	SSN	*Nautilus*	Electric Boat Division
572	52	SSR	*Sailfish*	Portsmouth Navshipyd
573	52	SSR	*Salmon*	Portsmouth Navshipyd
574	53	SSG	*Grayback*	Mare Island Navshipyd
575	53	SSN	*Seawolf*	Electric Boat Division
576	54	SS	*Darter*	Electric Boat Division
577	55	SSG	*Growler*	Portsmouth Navshipyd
578	55	SSN	*Skate*	Electric Boat Division
579	55	SSN	*Swordfish*	Portsmouth Navshipyd
580	56	SS	*Barbel*	Portsmouth Navshipyd
581	56	SS	Unnamed	Ingalls Shipbuilding Co.
582	56	SS	Unnamed	Open bid
583	56	SSN	*Sargo*	Mare Island Navshipyard
584	56	SSN	*Seadragon*	Portsmouth Navshipyd
585	56	SSN	*Skipjack*	Electric Boat Division
586	56	SSRN	*Triton*	Electric Boat Division
587	56	SSGN	Unnamed	Mare Island Navshipyd

Note: Six SSN's of the 585 design are included in the 1957 shipbuilding program before Congress.
Key: Letter N indicates nuclear powered; R indicates radar picket; G indicates guided missile.

WILHELM HADELER

THE AIRCRAFT CARRIER IN THE GERMAN NAVY 1934-1945

THE AUTHOR

a son of a public school teacher, was born on 12 November 1897, in Lauerburg on the Elbe. In January 1917, having finished his schooling in Lauerburg and Luneburg, he took his matriculation examinations and from the fall of 1920 he studied shipbuilding in the Technical University of Berlin—Charlottesburg. After completing his general examinations for the diploma in 1925, he stayed on at this university for the next 9 years as assistant to the Chair for Warship Construction. In April 1934 he became an employee of the Construction Office of the Supreme Command of the German Navy in order to carry out the preliminary work for the design and the building of aircraft carriers.

When the German-British Fleet Agreement was signed in the summer of 1935, he was entrusted with the design and the supervision of the building of the aircraft carrier "A."

After taking the master-builder examination, in the summer of 1939, directly before the outbreak of war, Hadeler was appointed to the Naval Construction Council. His more important wartime commands were: Division Chief in the Kiel and Wilhelmshaven Naval Shipyards; 1942-43 as Leader of the higher naval construction council in charge of naval construction supervision of the Deutsche Werke in Kiel; Managing Director of the arsenal in Gotenhefen; 1943-44 Advisor to the Supreme Command; and finally Instructor of Shipbuilding in the Naval War College in Flensburg-Mürwick. Presently, he is active in private industry.

Prior to World War II and since 1950, Hadeler has published numerous articles on the subjects of war-ship construction and the conduct of sea wars. These appear in various German technical journals.

LIKE THE *Flying Dutchman*, many a ship during World War II was veiled in mystery, but in most cases, details about them became known after the war ended. Only one large ship, the German aircraft carrier, *Graf Zeppelin*, remains a mystery. While much technical data regarding her has gradually become known, her fate has remained undetermined. Even as late as 1952, newspapers carried sensational stories about the German carrier.

But what actually happened? What did the Germans accomplish with this ship? What was her story in the various stages of planning, design, and finally, construction? How did the German Navy try to solve the many technical problems involved in carrier construction without any previous experience in this field? And, in the end, what happened to the ship, when, uncompleted and heavily damaged, she fell into Soviet hands?

In the following pages, the author, who was entrusted by the German Navy Command with the job of designing the first and only German carrier, reports on that monumental assignment, including execution of the design and the ship's construction. He supervised the actual construction from the K-Amt of the OKM (Construction Office of the Supreme Command of the German Navy). The follow-

ing report reveals a chapter of German naval history entirely different from what has been assumed, or reported by propaganda sources.

The construction of the carrier followed, logically, the general development of the German Navy after the first World War. When the German Navy, in the spring of 1920, began to rebuild what the war and its consequences had destroyed, it was confronted, materially speaking, with nothing but hulks.

The naval command, clearly recognizing the importance of a navy to a nation, courageously began its work. Obviously it was necessary to cultivate and maintain a feeling of "sea-mindedness" and to keep abreast of military and technical developments.

Actual operational planning had to be very simple and modest in view of the few remaining old ships, the lack of trained personnel and the general political situation. The scope of planning had to be limited to (a) defense of commerce in the Baltic against Polish forces possibly acting under French protection; (b) defense of German sea lines of communication in a commerce war presumably against the French. Such planning was completely Utopian, however, because even with modernization, existing German ships were not equal to these tasks.

The situation was not much improved even with the commissioning of the newly constructed 800-ton torpedo boats and the 6,000-ton cruisers. Even the completion of the three pocket battleships of the *Deutschland* class did not alter the situation very much; reports in the foreign press on the presumed threat of the German fleet were highly exaggerated. Compared with the fleets of the great naval powers such as Great Britain, the United States, France, Japan and Italy, in the period between 1930 and 1935, German "sea power" was negligible.

By 1930, the German Navy began to suspect that the chains imposed by the Treaty of Versailles might be eased, and eventually even shaken off entirely. It seemed that such a day was not too far off, even though no one had any idea that actually it was to come fairly soon. But when that day came, if it should, it was the duty of the German Navy to be prepared to accomplish, systematically and quickly, what heretofore it had been forbidden.

To be properly prepared, there was a particular need for mastering the most modern techniques in naval construction, as well as in naval engineering and armament design. Leaving the question of armament for the moment, it can be said that the German Navy made significant contributions in the field of naval construction such as introduction of electric welding, use of special steels, an improved system of static calculation, and, finally, introduction of sea waterproof light metal alloy. Much of the above resulted from the intensive research and development for the construction of the cruiser *Emden III*.

The same holds true in the development of light and high speed diesel engines for large ships, first used in the pocket battleships of the *Deutschland* class. All this required a knowledge, extensive at least in theory, of all types of ships and armament forbidden by the Treaty of Versailles. In general, experts had to rely on the experience of the Imperial Navy of the period before and during World War I, especially for information on battleships, heavy cruisers, destroyers and submarines. By following developments abroad, the Germans were able to utilize the knowledge so gathered when reconstruction was again possible. However, there was no starting point whatsoever for a naval air force and construction of aircraft carriers.

While the German Navy secretly had acquired a few planes around 1930 and had trained a few of its officers as naval fliers, this nucleus of a naval air arm, despite the enthusiasm and great vitality of its members, was unable to furnish any starting point in the completely unknown field of carrier aeronautics.

Work on developing the German Army started very soon after January 30, 1933, the day of Hitler's appointment as Chancellor. Because of Hitler's views and experiences, as well as his political plans, the bulk of the armament effort was directed toward the Army and the Air Force, even though he had officially promised to develop the Navy. Obviously rebuilding a navy is more time-consuming and also more costly than for the other branches of the armed forces.[1]

By the end of 1933, the Navy began, in addition to numerous jobs already underway, to consider the military and technical questions pertaining to construction of a carrier. Until then, the scarcity of officers and a multitude of other problems had interfered with even considering the carrier and its various aspects. Information from abroad on carriers had not even been collected or studied. Beginning with the theoretical side, planners concentrated on desk work, which, by the way, was not forbidden.

But it was discovered immediately that little could be accomplished in military planning without exact technical blueprints. We soon recognized that the difficulties involved were much greater than were at first anticipated. The German Navy had to begin in a complete vacuum. The accomplishments of the Imperial Navy between 1914 and 1917 were merely a point of departure from a period of a very honorable but also a very modest military tradition.

Before 1914 and in the beginning of World War I, the German Navy had relied largely on dirigibles of the *Zeppelin* and *Schütte-Lanz* type. These were far superior to contemporary naval aircraft of the 1912–13 period, although they undoubtedly had great shortcomings, such as their large size, extremely dangerous hydrogen filling and limited elasticity. When naval aviation started building in 1913 and 1914, naval airplanes relied exclusively on coastal airports on the shores of the North and Baltic Seas. So-called *Flugzeugmutterschiffe* (aircraft mooring ships) constructed after the outbreak of the war were nothing but small and easily movable bases for a very small number of seaplanes.[2] The old light cruiser *Stuttgart*, which had been equipped with a

light hangar on the stern, in the spring of 1918, held no promise of further development.[3] Plans for rebuilding the armored cruiser *Roon*, which was unfit for combat duty, remained unexecuted, along with the proposal to convert an unfinished liner at the Blohm and Voss shipyard at Hamburg into a carrier. So it was impossible to find any starting point whatsoever for planning a modern carrier.

But the foreign navies, as well, had almost no modern carriers, designed and built as such, which could be used as models. Even England, as well as the United States and Japan, had just reached the point where they could start with a completely new carrier design. Up to that time, whatever carriers there were, had been converted from existing hull types.

It was obvious that England has made the greatest progress. Following the *Argus* and the *Eagle*, the *Furious* had worked out well, militarily and technically, after major alterations resulting in a flush deck without an island. The sisterships, *Glorious* and *Courageous*, converted later, were considered by many experts as the best available aircraft carriers in 1933. In the United States, the collier *Jupiter* had become the experimental carrier *Langley*. On the basis of these experiments, the new battleships *Lexington* and *Saratoga*, still under construction in their docks and uncompleted because of the Washington Agreements, were converted to carriers. Doubtless they were a good piece of work for that period, probably even better than was imagined then. Yet to many experts, they were discouraging because of their large, if not too large, size for that particular time, and because they were considered somewhat unwieldy in their handling.

Unwillingly they gave support to the argument of carrier opponents, most of whom considered such large ships a mistake because they represented "too many eggs in one basket." They also argued that the protection of these "white elephants" among the battleships, would present a task to the fleet commander greater than all his other tasks, both general and special.

Although the British *Hermes* and the Japanese *Hosho* ships had been designed originally as carriers, the Germans knew too little about them, especially of the latter, to use them as models. Furthermore, the *Hermes* had been designed in 1917 when it was not even known whether the carrier plane of the future would have wheels or floats. Also, this was a period when experiments had been made only with take-offs from the ship; they did not know whether a landing on board would be at all feasible.

And finally, there did not exist at this time any mature or even clear concept on the general subject of carrier operations. How many times the plans of the *Hermes* were changed before her launching in September, 1919, or how many alterations were made before her completion, are still unknown. Altogether, her military value was so unproven that that ship could not be used as a model. Furthermore, what little knowledge the Germans had of the only new construction modern carrier, the USS *Ranger*, was of little value to them.

Germany then had practically no knowledge nor any detailed literature on aircraft carriers written by experts for experts. The best information available came from a lecture which had been given by the then Chief of the Bureau of Construction and Repair, Rear Admiral George H. Rock, before the Society of Naval Architects and Marine Engineers in 1928. The subject of that lecture was "Some Experiences in Connection with the Design of Aircraft Carriers and Remarks on the *Lexington* and *Saratoga*."[4]

In that paper, the main problems were described in military and technical terms. When the German Navy began to use this information, it was six years old and had become outdated in its factual details and even in some of its general principles. Nevertheless, the paper was considered an excellent introduction to the new field because of its emphasis on all the important details of the building program. Additional valuable material was taken from various articles which had appeared since 1931 in the United States Naval Institute *Proceedings* and in British magazines.[5]

From the above, the following broad specifications were evolved:

> For study purposes, an aircraft carrier is to be designed to displace 25-30,000 tons; to carry 50 to 60 planes; to have a speed of 33 knots and an armament of eight 20.3 cm guns in four twin mounts. Strong anti-aircraft armament is to be provided.
>
> Armor protection is to be on the same scale as that on German cruisers.

It was pointed out immediately that these specifications were for planning purposes only. The author learned some years after the war, that they had been drawn up without any conception of the operational or tactical employment of a carrier. The German Admiralty, on the basis of Weyer's *Taschenbuch der Kriegsflotten* and Jane's *Fighting Ships*, had outlined what it thought it could ask for.

The above in some respects simplified the job for the author as the designing shipbuilder, but in other respects, made his job more difficult. It simplified the task because the specifications were broad and as a result the question of resolving the technical and military incompatibilities presented no problem in the early stages of the planning. The problem was further eased by the fact that this was a design study only. However, it was more difficult because no one in the German Navy knew what qualities an aircraft carrier should and must possess. The prescribed specifications might, after all, be completely incompatible.

Specifically, no one knew the relationship of the aircraft to the guns. Which was to be the main battery of the ship? Were the aircraft and guns to be equally important? Altogether, what would be the relationship of the guns to the aircraft?

This is why the preliminary planning for, and the design itself of Carrier "A" (the name *Graf Zeppelin* was, in accordance with German custom, given at the time of launching) was the only time in German naval history when technical considerations controlled the planned combat capabilities of a large warship.

The work itself began with collecting, organizing and using everything found in such publications as Weyer's *Taschenbuch der Kriegsflotten* and Jane's *Fighting Ships*, as well as in available articles, reports, monographs and pictures of all kinds from the domestic and foreign daily and technical press. The importance of this collection was that it revealed, to a certain degree, the concepts of the large navies as to the general characteristics of carriers. We gathered from this information that the carrier (and its planes) was still generally considered a complement to the "classic" arms—the gun and torpedo. Its major task was considered to be reconnaissance.

But how could we start the design? To begin with, we had to decide whether the ship was to be a flush-deck type or would possess an island. This was a controversial question, although the leading foreign navies had decided on the island type some time earlier. So had the author from the very beginning, using the British *Courageous* class as a convincing argument. The decisive reason, however, lay in the fact that the United States Navy had designed its *Ranger* with a flush deck, but later gave it an island super structure, although only a small one. These reasons were accepted, and so the German decision was in favor of the island type.

One thing which had a fundamental influence during the planning stage (and it might appear well-nigh incredible to both domestic and foreign experts) should be emphasized at this point: *The whole task was begun and in the early stages executed without any participation on the part of the Luftwaffe.* This resulted in some insurmountable difficulties, but it also had its advantages.

This lack of participation in planning by the Luftwaffe can be explained by the political conditions which existed immediately following 1933 and especially by the fact that the Treaty of Versailles had barred any military aviation. Officially there was no German *Luftwaffe* in the spring of 1934.

Because of the limited personnel and material available to the clandestinely organized Luftwaffe, no one in that organization would concern themselves with questions dealing with aircraft carriers and carrier aircraft. Another aspect was that Goering, as Minister of Aviation, fought for an independent Luftwaffe which would contain all German military aviation. Under these circumstances, the growing Air Force had no interest in a naval carrier project, especially since it was merely a project for study only. There were, however, officers with a naval background who were eager to cooperate. In fact, their idea of a naval air force, under Goering's command, with the carrier playing the principal part, was discussed on several occasions.[6]

But because of strict security, it was impossible to use whatever personal connections existed between the author (in his capacity as shipbuilding expert of the project) and individuals in the technical office of the Air Ministry. The general restraint shown by the Air Force representatives was officially excused by the lack of personnel in the *Luftwaffe* as well as in the aircraft industry entrusted with actual development of the air force.

All this made the job very difficult. In restrospect, it can be given as the reason why the German aircraft carrier in its general construction was not as successful as it could have been, and why it did not show the balance between the offensive and defensive capabilities which as a rule was self-evident in all German naval construction.

But as we have said, the absence of the *Luftwaffe* resulted in some advantages which should not be underestimated. There was, for instance, no disagreement as to need for the island, as was the case in the British Navy during the period from 1919 to 1922. Furthermore, the size of the aviation spaces could be determined without arguing over every inch of the length of hangars and flight deck. Hence much time was saved. It seems preposterous, nevertheless, that the most important of all questions regarding carriers was not resolved at first: the size and probable weight of the planes to be used on the ships. The Luftwaffe's fear of becoming committed to design of planes for this ship was the reason for that.

Then what could we do? A design without such data was impossible; yet a shipbuilder must find a solution in such a situation, even if he can only guess at the missing numerical values. Therefore the author sifted from the collected literature, all the information of length, width and height of British, American and French carrier planes. He also learned about types with folding wings and how much they would decrease space requirements, and finally, the weight of each of the different types of planes. Once the highest possible numerical values had been determined, we added fifteen to twenty percent as a safety factor. Thus the basis for the design was prepared. After obtaining this information, we began estimating measurements of elevator platforms, width and height of hangars, strength of the flight deck, etc. Later these figures were found to be thoroughly reliable (at least during the period when the carrier was still under discussion in the German Navy), particularly since the strength requirements which originally were measured too generously, later proved a most welcome reserve.

It was in the early summer of 1935, when the study project entered the phase of serious design, that the astonished Luftwaffe experts asked who had established the definite specifications. In their view the specifications were inadequate, but they became reconciled very quickly when the foregoing details were explained.

It is obvious that under such conditions the many specific problems pertaining to flight-deck arrangements, such as take-off and landing areas, arresting gear, wind protection screens, etc., presented many problems. Such also was the case with the details of fueling and arming the planes within the hangar and on the flight deck. Difficult also were such considerations as installations for moving and anchoring the planes on deck and in hangars, and, of course, fire protection. The author knew nothing of aviation and had never flown, not even in a commercial plane.

There was no pilot in Germany with any carrier experience to advise us. But with more and more research and by making the most of what information we had, plus a very thorough study of seemingly irrelevant news reports and pictures, we were making headway, but primarily in theory only. We found consolation in the thought that, once the construction phase had been reached, it was up to the shipbuilder to translate theory into practice.

Very soon after starting the project, while still collecting information, the author tried to produce a general design but it turned out to be impossible. The usual rules of ship design, such as those used in the estimation of displacement, were of no help in the case of a carrier. The construction of such a type, once external conditions such as the measurement of canals and locks are disregarded, is mainly a question of space for the aviation department particularly length and width of the flight deck and size of hangars. This was the first and most important discovery, but it took time before all participating agencies would agree to it.

It soon was obvious that the armament originally planned—eight 20.3 cm guns in four twin mounts—could not be placed in any satisfactory manner. When the author explained, at a conference of the participating agencies, why this was so and added that this caliber was placed on American and Japanese carriers at the time of the Washington Conference for political rather than military reasons, it was agreed to omit them. At first we agreed on eight 15 cm guns, but asked that to guarantee sufficient defensive power against cruisers and destroyers, their placement should provide for four guns bearing in every direction. This could be accomplished by placing the guns, preferably, in recesses in the ship's side, as the British had done in the *Courageous* class.

All in all, those particular ships served, in the broadest sense, as the principal models. Their external and internal structure, as well as their offensive and defensive capabilities, fulfilled in many respects the German requirements, or at least tended to do so. Besides, the most complete information we had was on these ships—a factor which under the circumstances carried great weight!

Further experimentation led to a phase in which these British ships were copied as closely as possible. This offered an opportunity to discuss the specifications in details. In each case the questions were: What is the purpose? How does it function? Why is it shaped and constructed in this particular form? Simultaneously, while seeking the answers, we attempted in each case to utilize any specific experience of the German Navy and thus to find a "translation into German," which would be followed by a truly German solution, corresponding as much as possible to German ideas and requirements.

The military were kept informed on the general state of affairs and on newly acquired knowledge. Their reactions to any given, or even possible, solution often resulted in valuable suggestions, so that this exchange of ideas revealed more clearly the desired specifications. Required speed was increased to 35 knots soon after the work was started. It was decided in the early planning stages that carriers probably would have the largest engineering plant so far used in any warship, with the exception of the USS *Lexington* and *Saratoga*. As early as the fall of 1934, we anticipated a requirement of 160,000 to 180,000 shaft horsepower, and the final design was executed with 200,000.

Only a high pressure turbine plant could produce such power, with a steam pressure of sixty or seventy kg/cm and corresponding temperatures. Such an installation was then being prepared for the battleships of the *Scharnhorst* class. For use on the carrier, its power output was increased to the limit for the following reasons: it was not impossible that the ship's displacement might become considerably greater during construction than was originally planned; details of many installations, especially for the aviation department (elevators, arresting gear and so on) were totally unknown and could not therefore be estimated as to their possible weight. It was anticipated, however, that the speed—a factor of the greatest importance to the German High Command—would not go below 35 knots despite a possible increase in displacement.

By the spring of 1935, the situation was clear enough for us to prepare, on the basis of military requirements and in a relatively short time, a design which could serve as a basic construction plan. A need for that developed sooner than had been anticipated. On June 18, 1935, the German-British Fleet Agreement was signed. This meant that, politically, Great Britain had renounced officially those sections of the Treaty of Versailles which limited naval rearmament; thus the Navy finally had been given the right to build its ships according to its own plans and ideas. A few weeks later, after the design for a carrier with a standard displacement of 19,250 tons was completed, the *Deutsche Werke Kiel Atkiensgesellschaft* (DWK) at Kiel was entrusted with her construction. Also during the same year, the same shipyard received an order for the construction of a ship officially named "Aircraft Carrier A." The yard called it "K 252" (K standing for Kriegschiff (battleship) or possibly Konstruktion (construction)) because it was the yard's 252nd new construction vessel.

The information which was given to the yard and also to the public at that time, correspond to the plans, with the exception of speed:

Standard displacement	19,250 tons
Length of waterline	250 meters
Width of frame	27 meters
Draught at 19,250 tons	5.6 meters
Armament	Sixteen 15 cm guns Ten 10.5 cm AA guns Twenty-two 3.7 cm machine guns and also some 2 cm machine guns
Aircraft	About 40

Speed was described as 32 knots based on 120,000 shaft horsepower. As there was no limit on the speed, nor contractual arrangement regarding it, this information could not be considered as misrepresentation.

Intensive work, which was started early in the summer of 1935, finally resulted in the necessary collaboration with the *Luftwaffe*. In fact, from 1936 on, it was a very close and successful one, largely because of the appointment of a liaison officer who was an experienced and well-trained ship-building engineer. During the first World War he had distinguished himself as a naval aviator at the Dardenelles and later was active in the new *Luftwaffe*, particularly in the field of ground organization.

Among the above mentioned data, the sixteen 15 cm guns for naval armament might be of special interest to the reader since originally only eight had been discussed. This heavy armament caused much speculation and convinced foreign experts like Pierre Barjot and Raymond V. B. Blackman and others, that the German Navy was attempting, with the creation of this new carrier, a new type of ship combining cruiser and carrier, with the emphasis however on the latter. There was also speculation on her employment as a commerce raider.

In reality, things were different—and much simpler. It was because of the original and complete absence of the Luftwaffe in the early stages of design and preparation, that the Navy considered aviation problems only on the basis of its own experience. On the other hand, too much importance was attached to the guns. None of the responsible naval officers believed that such a large ship could have sufficient offensive power without relatively effective armament. That was indicated by the original plan for eight 15 cm guns.

The reason for the increase to sixteen guns was simple: as early as the fall of 1934 it had become obvious that it would be difficult to find sufficient space on the stern for placement of two 15 cm guns on both sides in addition to the hangars, whose width had to be determined by the size of the planes. Finally, their placement was planned in recesses in the ship's side, as in the case of the *Courageous*. The plan was to use center-pivotal gun mounts with protective armor. To simplify matters, the author proposed combining the two guns on each side in a twin gun-carriage with the same protective armor as had been planned (but not executed) for the light cruiser *Emden III*. But after long discussion, on the suggestion of an enthusiastic gunnery expert, a new solution was found, not corresponding at all with the character of the ship. Fore and aft, an armored casemate with two twin gun-mounts was built in on each side, despite the impossibility of stowing therein a sufficient amount of ammunition for the caliber and of guaranteeing a complete crew for this kind of gun.

While by the summer of 1935 the general problem of the design had more or less been solved, there was still a definite uncertainty regarding technical details, other than those already known from studying existing types of ships. For this reason the Navy sent one of its leading construction officials (who was also the author's superior in organizing the work) to Navy Week in England, when it was learned that a carrier could be inspected. Results of this visit were unsatisfactory, however, because nothing essential was revealed. We were delighted therefore when the Imperial Japanese Navy, after some negotiations, declared its readiness to show one of its carriers to a group of experts and to demonstrate her in operation.

An Air Force officer, a naval officer, and the same official who had gone to England, went to the Far East in the fall of 1935. They saw a few things, but all told, this trip likewise did not yield the anticipated results. The "Japan Commission," as the Navy unofficially called it, returned with a voluminous report and about 100 detailed blueprints of the carrier *Akagi*, which led to the persistent rumor that the German carrier was based on Japanese plans. In reality, their influence on the plans for "Carrier A" was quite negligible. The most important change made as a result of the Japanese plans was provision for a third elevator. The plan for a forward launching area for fighters, which originally had been drawn in accordance with the plans for *Courageous* class, had been abandoned because there was not sufficient runway for the BF 109 (Messerschmidt fighter M-109).

The real significance of the Japanese report and blueprints was that they proved the work of the K-Amt of the German Navy had been right. That in itself was a success, considering the difficulties and circumstances described earlier. Also, the blueprints offered valuable aid and suggestions, in some instances, to further developments. In particular, the design of the arresting gear with brake cables was taken from the Japanese designs. But on the whole, the German Navy had to rely on its own ingenuity and efforts.

With the spring of 1936, work became more intensive. By this time, all the Supreme Command's experts on technical and military details were participating. Under guidance of the shipyard, firms qualified for handling special installations were consulted: and, on the basis of close cooperation between *Luftwaffe*, Navy, shipyard, and special firms

we began with a far-reaching experimental and development activity designed to complete the special installations for the aviation department.

The keel of carrier "A" was laid in dry dock I of the DWK at Kiel on December 28, 1936. Three weeks later, the battleship *Gneisenau* was launched. As the latter approached completion, the carrier came to be the main construction project of the yard, which also was busy with construction of the supply ship *Franken* and of submarines. The German Navy's first carrier was launched on December 8, 1938. Hitler, Goering, and numerous high ranking officers of the Navy and *Luftwaffe* were present. On Hitler's orders, the carrier was christened the *Graf Zeppelin* by Countess Hella von Brandenstein-Zeppelin, daughter of the famous airship builder. None of those present on that light December morning suspected that the proud ship would never be completed and would come to an inglorious end.

Meantime, the neighboring *Friedrich Krupp Germania-Werft* received orders to build a sister ship, the aircraft carrier "B." She was to be identical with the DWK construction. But progress on this second ship had been slow, so that only a few hundred tons of material had been assembled at the time of the launching of carrier "A." The only outstanding innovation in her construction, was that she was to be launched with her bow lacking a 30-meter piece. This was necessary because the length of the free water outside the dock was not sufficient to permit the launching of a complete ship. But even launching a "shortened" carrier presented considerable difficulty and required extensive experimentation with the brake installations to be employed immediately after the ship hit the water. This was necessary to avoid any damage to the ship from the opposite bank. It was planned to add the missing part later in the dock. Because of this difficulty, transfer of the construction to the DWK had been contemplated for some time, even if it were necessary to sacrifice a considerable portion of the already used materials. Nothing was ever done about it, however.

During 1939, construction on both ships proceeded according to schedule. The *Luftwaffe* at that time began preparations for training the first carrier squadrons. In this connection, an experimental apparatus with brake installation at the *Luftwaffe's* test station at Travemuende played an important role. The planes intended for use aboard the carriers were not, as was the case with the large navies, a type which had been developed for carrier use. The *Luftwaffe* was unable to offer anything in the way of design or production of carrier planes.

In fact, the Air Force had concentrated on using models, already in the mass production stage, with added improvements and changes for carrier use. Provided for use as fighters were the already very successful BF 109's, and for dive-bombers, the JU 87, for which folding wings had been developed. At a rather early stage in the planning, a type suggested for torpedo launching was eliminated since the JU 87 could be so used without extensive changes.

In the first months of planning and even of actual construction, we planned to use propeller-type planes, but by the middle of 1937 the need for a catapult to start the planes, at least for the fighters, became obvious. For this reason, two DKW-built catapults with compressed air motors were built into the fore part of the flight-deck. Thus it was possible to allow take-off of one plane per minute, and by interchanging start, sixteen planes could take off in seven and a half minutes. Obviously, such a performance would require highly trained personnel. The tasks such as starting the engines, aligning and securing the aircraft on the catapults and returning the catapult carriages after each shot were not easy ones. But the installations were finally completely planned and tested, as far as possible, with the existing testing facilities.

Unfortunately, a plan for replacing the starting process by compressed air, which was space consuming and very heavy, by one of concentrated hydrogen peroxide, was given up immediately after the war broke out. This might have meant that catapulting all planes on board would have been feasible in a very short time, particularly since it had been decided in the meantime to catapult all carrier planes, including the torpedo-carrying JU 87's.

The outbreak of war on September 1, 1939 found the Navy still being developed and far short of what was considered necessary for any successful sea warfare. The aircraft carrier had progressed only to the point where it was anticipated, at least with normal construction conditions, that she would be commissioned by the end of 1940. But now it was necessary to concentrate all our energy on submarine warfare. Any work on large surface vessels was possible only as a fill-in activity. This resulted in increasing delays and finally, by May of 1940, all work on the carrier hull was stopped. Any hope for the timely use of the ship in battle was abandoned. By the beginning of the next month, in order to provide berthing space and to remove the carrier from possible air attack, the ship was moved to Gotenhafen (Gdynia). Her naval gun installations were removed and put to use somewhere in Norway for coastal defense. The anti-aircraft guns on board had already been used occasionally for defense against enemy aircraft. It was on one of these occasions that the ship received its first hit: the anti-aircraft guns then had no range limits and in the heat of battle, a 3.7 cm shell hit the slanting radio mast which was attached to the smoke stack. Because of the close proximity of the gun to the island, the missile had penetrated the mast without exploding. Carrier "B," which had progressed to the point where the first plates of the armored deck were being installed, was broken down on the launching ways to provide space for submarine construction.

The High Command occasionally did some work on carrier projects during the first years of the war, utilizing ideas from the captured plans of the French carriers *Joffre* and *Painleve*, but there were no defi-

nite results. In 1941 some ideas on a hybrid carrier were considered carefully; in addition to other proposals, a heavily gunned battleship with a flight-deck and hangars was suggested.

This technical aberration, as this plan after sober consideration can only be called, apparently was based, in the final analysis, on a remark of Hitler's. He had mentioned shortly before the launching of the *Graf Zeppelin*, in a speech on technical and military problems of the carrier, that such a vessel with sufficiently strong armament should be excellent in warfare against enemy shipping. None of the influential persons involved knew then, that the hybrid cruiser was regarded by the British Navy as early as 1918 and 1919 as impossible, or that the *Vindictive* had been found a complete failure during construction.

In any case, carrier plans were taken up again in the spring of 1942. Destruction of the *Bismarck* and the part played in it by British aircraft carriers, as well as the general development of sea warfare, underlined the significance of the carrier as a type of vessel so that finally the Supreme Command yielded to the Naval Command. Not only did the *Luftwaffe* approve completion of the *Graf Zeppelin* and begin to train carrier squadrons, but it was even decided to convert to carriers the heavy cruiser *Seydlitz* which was nearing completion at the *Deschimag (Deutsche Schiffsund Maschinenbau A.G., Werft A.G. Weser)*, and also the fast *Norddeutscher Lloyd* liners, *Potsdam, Gneisenau* and *Europa*. But a complete study soon revealed that the passenger liners would not prove satisfactory due to lack of general stability. Thus their conversion never progressed beyond the idea stage. But the *Seydlitz* was completely "shaved," viz. had lost its superstructure by February, 1943, when construction of all large ships was stopped. The author does not know whether any other preparations for conversion to carriers were undertaken.

In the process of construction on the *Graf Zeppelin* before the outbreak of war, the weight of the island had been increased considerably so that the resulting heeling could be avoided only by radical measures. The reason for the heeling was that instead of the relatively light guns originally prescribed for the heavy anti-aircraft battery, very heavily armored guns finally had been used. Instead of a light chart room, an armored bridge with an armored chart room had been built; and the light masts originally planned for the antennas had been replaced by heavier ones. Sufficient fuel storage tanks were not provided, as in the beginning none had been requested: now it was ordered that blisters be added to the ship. It was hoped that they might improve protection against torpedoes. By building them of 20 mm plate on the port side and of 10 mm plate on the starboard side, they served to balance the increased weight of the island. Furthermore, certain details of the ship had to be changed as the result of progress made by the *Luftwaffe*, and experience gained during the war.

The continuing construction orders were given to the *Deutsche Werke Kiel A.G.* at Kiel. The author, who had entered the Navy on active duty as *Marinebaurat* (naval construction advisor) and after the outbreak of war was ordered from the OKM to the naval shipyard at Kiel and later at Wilhelmshaven, was entrusted with general supervision of construction. (Since 1938 he had had almost no connection with the vessel.) Moving the ship, then at Stettin, was delayed by the authorities from summer until late fall, as they feared enemy aircraft attacks, especially when under tow. At last the ship arrived at Kiel on a foggy December morning. While anchoring in the yard canal, the carrier nearly upset a derrick on the shore. This was due to her "Atlantic Bow," as the over-hanging clipper stem which, after launching, had replaced the original rather steep and straight stem, was called. Superstitious yard workers saw in this a foreboding of bad luck for the ship!

A few days later the huge vessel, with a height of 22.5 meters up to her flight deck, was secured in the yard's 50,000 ton dry-dock. The ship's underwater structure was in perfect condition, although the vessel had not been in dock for three years. Work on the pre-fabricated blisters could be started immediately. The final objective had become a very modest one: it was now hoped that the carrier could be completed with the two inner shafts of the engineering plant and thus be available for trials, so that some much needed experience could be gathered and the first squadrons receive training on deck.

Yet things turned out differently. On January 30, 1943, Great Admiral Doenitz replaced Great Admiral Rader. Orders to stop all construction on large vessels were issued. Construction on the *Graf Zeppelin* was completed to the extent that in case of emergency, she could be navigated and kept afloat with the help of pump-ships. In March, 1943 she was again moved to the East and placed in the Moelln-Fahrt, an arm of the Oder river to the *Dammsche See*, a part of the mouth of the Oder *(Stettiner Haff)*. A small crew of shipyard workers remained aboard for the most elementary maintenance and preservation. In the summer of 1944 the possibility of using her as living quarters for cadets and midshipmen was considered. It was in the course of an inspection trip in connection with this proposal, that the author, who in the meantime had returned as *Referent* (inspector) to the OKM, saw the ship for the last time.

It was a painful reunion, for the ship, now completely covered by camouflage nets, had suffered through lack of maintenance due to increasing scarcities. In the course of time, part of the auxiliary machinery had been removed and used as spare parts for similar engine plants on battleships. Furthermore, it was obvious to the author that the carrier would never be completed, even in the event of

Figure 1. *Graf Zepplin* before launching.

a not too unfavorable end to the war (by that time practically impossible), because her general plan was thoroughly outdated.

The end came soon. When the Russians were approaching the Oder, orders were given to "scuttle" the ship, which we could no longer tow away, by tearing out all the important stop-valves in the pump ducts. This action, it was anticipated, would complicate tremendously the enemy's towing her away, since a ship thus stripped would sink, even with a most insignificant leak. The order was executed and the torn-out parts were sent to the West.

But the final catastrophe could no longer be put off. Soviet troops had reached the right bank of the *Oder* and the island of *Wollin*. *Graf Zeppelin* had been hit several times by artillery and was abandoned by her crew. On orders of the senior naval officer in the *Stettin* area, depth charges were placed in her boiler and engine rooms. They exploded exactly at the prearranged time. The explosion was marked by the smoke clouds issuing from her smokestack. Not only did it render the entire engine plant useless, but in all probability caused such leakage that the ship finally sank.

However, there were only 50 cm water under the keel, so the carrier could not heel over but settled down in the mud. It was not, therefore, a difficult job for the Soviet salvage force, in the period from 1945 to 1946, to stop the leaks and move the vessel to *Swinemuende*. She was seen there, beyond a doubt. While moving the ship to Leningrad, the Soviets used the hangars as stowage space for booty, mainly machinery. The vessel departed in 1947, towed by heavy Soviet tug-boats. It is quite likely that she never reached Leningrad, as no reliable reports on her arrival were ever received. A ship 262 meters long and with about 18 meters free-board cannot be hidden throughout the years, even by the Soviets, without some information leaking out. Of the many rumors and alarming reports about the ship's real end, one from Sweden seems to be the most probable. According to this report, the ship hit a mine to the northeast of Ruegen and sank. The heavily loaded hangars would probably have changed the ship's balance because of the shifting

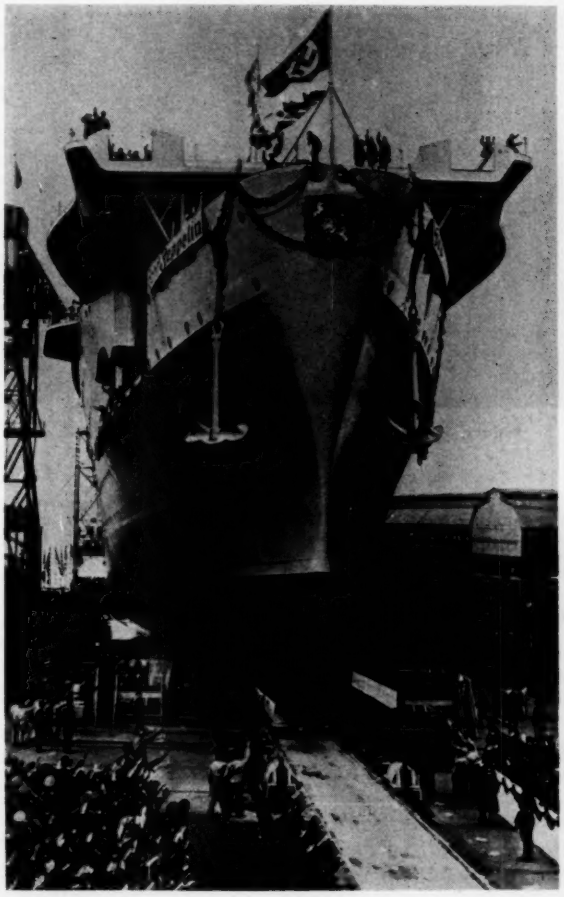


Figure 2. Launching of German Aircraft Carrier *Graf Zepplin*.

of gravity toward the top, so that the ship must have keeled over quickly. Only time will tell whether the location of the catastrophe ever will be known.

Sensational news stories and repeated reports on the existence and dangerous utilization of the ship as a carrier and launching station for guided missiles were nothing but propaganda. To the expert, such reports are out of the question. But if by some chance, contrary to any expectation, the vessel should still be harbored in some small and distant bay of the Finnish Gulf, she would be so outdated and corroded that her reconstruction for any purpose would not pay off the Soviets. The engineering plant could not be completed without the detailed plans, or without the highly qualified experts and capabilities of *pre-war* German industry.

Count Ferdinand von Zeppelin, builder of the famous dirigibles, died one of the great in the realm of technical achievement. Yet the ship that bore his name remained, despite the mental and material efforts of all involved, nothing but an insignificant episode in the background of the greatest of all wars.

Figure 3. Launch of the *Graf Zepplin* completed.

Figure 4. *Graf Zepplin* after launching.

FOOTNOTES

[1]More details may be found in a booklet which is little known in Germany and not at all abroad. It was written by Hitler's naval aide, K. J. von Puttkamer, Die *unheimliche See* (The Frightening Sea) (Hitler und die Kriegsmarine—Hitler and the Navy). Publisher Karl Kuehne, Vienna-Munich.

[2]The following mooring ships were commissioned, some of them for only a short time:

Answald	5401 br tons	11 knots	3 planes
Santa Elena	7415 br tons	11 knots	4 planes
Oswald former *Oswestry*	5401 br tons	10 knots	4 (?) planes
Glyndwr	2425 br tons	9 knots	(no hangar)

[3]*Stuttgart*, 3469 metric tons; length 116.8; width 13.3; draught 5.4 m; 12.000 PSi; 23 knots; 4120 sea miles at 12 knots; 10 of 10.5 cm; two planes in hangar; in case of emergency one space available on the quarterdeck.

[4]*Transactions of the Society of Naval Architects and Marine Engineers*, vol. 36 (1928), p. 63.

[5]As the most significant may be quoted:
H. B. Miller, "Covered Wagons of the Sea" in *U. S. Nav. Inst. Proceedings*, vol. 57 (1931), p. 1461.
M. Prendergast, "Wings and Wheels, Wood and Wire." (British Aircraft Carriers 1914-1924) *Ibid.* vol. 58 (1932), p. 205.
W. R. D. Acland, "Deck Flying." *Jounal of the Royal Aeronautical Society*, vol. 35 (1931), p. 273.
W. R. D. Acland, "The Development of the Fleet Air Arm," in *Journal of the Royal United Services Institution*, vol. 79 (1934), p. 260.

[6]Compare Edward L. Barker, "German Naval Aviation," *U. S. Naval Inst.*, vol. 76 (1950), p. 731.

REAR ADMIRAL H. G. RICKOVER, USN

METALLURGY IN ATOMIC POWER

ACKNOWLEDGEMENT

This is a speech which was delivered by Rear Admiral Hyman G. Rickover at the National Metals Congress of the American Society for Metals at Philadelphia, Pa. on 20 October 1955.

CHAIRMAN STRAUSS of the Atomic Energy Commission was to have spoken to you tonight. He has asked me to express his regrets that he is not able to be here. Also, he has asked me to tell you that he recognizes the important role your organization plays in supporting and extending our country's atomic energy program.

I cannot give you the broad view of our program that Mr. Strauss would have presented. But I do appreciate this opportunity to speak to you because we have major problems in the use of metals in the application of atomic energy to the propulsion of ships and I am sure you can help solve them.

The successful operation of the *Nautilus* assures the application of nuclear power to many other naval vessels. Such application must be based on sure knowledge of all aspects of nuclear power plants, including materials performance.

Although I will discuss primarily the problems associated with naval plants, I am sure you will recognize that what I say applies with equal force to nuclear power plants for central stations, such as the one we are building at Shippingport, Pennsylvania.

The problems which face us are of two kinds. First, are those unique to atomic energy—such as the fuel elements and other reactor components that use uranium, zirconium or beryllium and which must withstand the impact of the nuclear fission process. Second, are the ones we have with conventional materials and components—such as carbon steels and stainless steels which are used in valves and heat exchangers; these are just as important as the special materials.

It is a common belief that atomic power development is primarily the province of the nuclear physicist. Nothing could be further from the truth. What actually faces us is how to determine, by calculation and by experiment, the best way to remove heat from a reactor, and then to design and build all of the components of the reactor system so that they will operate safely and reliably under the most rigorous conditions.

The problems include the familiar ones of using materials at high temperature and pressure. They also include the ability to withstand corrosion to a very high degree. The new problems raised by atomic power are caused by the fact that in nuclear fission heat is generated inside a fuel element and proceeds outward, whereas in the chemical combustion process the heat is applied from the outside and proceeds inward. In chemical combustion we are normally limited to a maximum of about 3000°F, but in nuclear fission we can achieve temperatures of millions of degrees. Moreover, we are faced with radioactivity and its tendency to distort and to weaken materials. The fact that many materials capture neutrons and thus tend to stop the chain reaction severely restricts us in their choice.

First of all, the development of suitable reactor fuel elements is the greatest problem we face. The fuel elements are those parts of the reactor core that contain the fissionable material which produces the heat, and hence must withstand high temperatures and the radiation impact of the fission process. Changes in form and substance in fuel elements are inherent in the operation of a reactor. And yet these changes must not be allowed to destroy the over-all integrity of the fuel element nor interfere with the functioning of a

complex reactor with its close tolerances. The service demands on fuel elements are greater than those I know of in any other application. Work on fuel elements is a challenging new field which is receiving wide attention, and in which millions of dollars are being spent. But much creative and fundamental work still remains to be done.

The problems in fuel elements involve new materials such as uranium, zirconium, and beryllium. These are well known to you. Your society has held technical sessions on both zirconium and beryllium, and we have worked together publishing a book on zirconium and one on beryllium.

A challenging problem in the design of the *Nautilus* reactor was the development of zirconium. It was selected in 1949 because it had a low cross section for the capture of neutrons and because it was also supposed to be corrosion resistant. I vividly recall my concern when I learned that this material which we considered extremely desirable was not in production on a commercial scale and that its engineering properties were, for all practical purposes, unknown. To be sure enough was known to indicate that it could be a suitable material, but you all know the vast gap between meager laboratory data and the full information which is needed for practical design.

We were faced with the problem of continuing which zirconium or of employing the "green pasture" approach—that is, changing to another metal. The "green pasture" concept is a very seductive one. You know the faults of what you are working with, but the new metal always appears to be much more attractive because you haven't yet discovered its faults and difficulties.

Fortunately we stuck with zirconium. The naval program would have suffered substantially had we abandoned zirconium at that time. We decided to tackle the difficulties and we set up a very tight schedule for solving them. A number of the organizations represented here tonight participated with us in the crash program which led to the investigation of its engineering properties and to its production in commercial quantities.

While the production problem was being solved we learned that zirconium had corrosion qualities unsuitable for our purpose. Again an integrated crash program between industry and government got underway and was successful in providing corrosion resistant metal in time.

A "task force" consisting of a representative from each organization doing work on zirconium was established to develop and select a more corrosion-resistant alloy. Each representative had access to his management so that the necessary changes in the program would be effected without delay. The result was development of Zircaloy-2 in time for its successful use in the *Nautilus*. Without this urgent need, and the concentrated attack, it probably would have taken many years before zirconium reached its present state of development.

Another example where management is helping the development of new materials is in the use of aluminum for cladding fuel elements. As you know, aluminum is much cheaper than zirconium, and has as good nuclear characteristics. The problem is that for power reactors, the temperatures are so high that they result in excessive corrosion of presently available aluminum alloys. One large aluminum producer is spending its own money, in cooperation with AEC laboratories, to find alloys which will be satisfactory. More companies should analyze future needs and push such developments.

The development of uranium, which is essential to nuclear reactors, has been marked from the outset by insufficient knowledge of its properties. This has made it extremely difficult to develop processes for its production. For instance, as late as 1942 the tables gave the melting point of uranium at 1800° centigrade. It is now known to be 1133° centigrade. And yet uranium has been known since 1789 and was in use for many years prior to 1942. The actual melting point only became known during early casting operations carried out for the atomic bomb project.

So far, the metallurgical problems that have arisen in the development of nuclear power have had to be solved by empirical means, since there is a scarcity of knowledge of the fundamental principles involved. Thus, our metallurgical advancements have required the use of extensive manpower and the expenditures of large sums of money. Until the fundamental processes involved are understood, additional large-scale use of manpower and money will be required to solve each slightly different problem as it arises. This empirical approach has just about reached its limit. The methods we are using are rapidly becoming unsuitable for solving our problems.

The empirical approach has also resulted in unexpected and unexplainable flaws and failures. These continue to plague us, and cause considerable delay.

Let me illustrate what happens when there is a lack of fundamental understanding of metallurgy. About a year ago we decided to do basic research in uranium, in the attempt to develop an alloy resistant to corrosion in water at high temperature. A number of the most experienced metallurgists and solid state physicists agreed to help. I asked them this question: "Using the phase diagrams and other characteristics of uranium alloy presently known, what particular region would you recommend as one which offers possible promise?" They replied that they didn't even know where to start. As a result, an extensive and costly empirical program has had to be undertaken.

Now, I would like to review some of the specific metallurgical problems we are facing. But some may say: "It is easy to list problems, but difficult to solve them." To this I would reply: "In achievement I expect to fail, but let me never recoil from endeavor."

A major problem we face in the application of nuclear power is that of corrosion. We like to use thin sections of protecting metal as a jacket between the radioactive fuel and the coolant which conducts the heat away. Thick sections of metal require the use of additional nuclear fuel. This tends to make the reactor larger, reduces the neutron economy, and re-

sults in increased cost. But the use of thin jacket sections of metal makes it more possible for radioactive fission products to escape into the coolant stream if undue corrosion should occur.

The undesirable effects of corrosion are not, to be sure, unique for nuclear power, but their effects can be more serious. The special coolants and operating conditions in nuclear power plants present corrosion problems quite different from the problems previously faced by industry. Most of our corrosion work has had to be done by testing a large number of slightly different materials under a wide variety of conditions before a satisfactory one could be found. Even a partial understanding of the mechanism of corrosion would permit a drastic reduction in the number of tests.

It has been estimated that corrosion of metals entails a waste of about four billion dollars per year. This points up the importance of launching a large-scale program to learn the mechanism of corrosion.

In addition to the problems of fuel elements and corrosion resistance of materials, a third area of concern is the application of conventional materials to nuclear reactors. Here also the lack of knowledge of fundamentals is holding us back.

In a large number of instances where conventional materials were specified for nuclear power plants on the basis that these materials had been used effectively and for a long time by industry we found that the basic knowledge of the reasons it had been possible to use the materials effectively was extremely meager.

In order to apply these materials to nuclear power plants with some degree of assurance that they would work satisfactorily it became necessary to make a large number of tests since there was no hope that an understanding of the fundamentals involved would be reached within any reasonable length of time. Of these conventional materials, stainless steel has been our biggest problem. This steel has many fine properties and will continue to be important in nuclear work. However, we are demanding a great deal from it. In addition to its use for structural members, applications include use for many parts with close tolerances which are lubricated by hot water.

The state of knowledge in the welding of stainless steel is far from satisfactory. It has been necessary to devote an extreme amount of attention to its fabrication and welding, to assure that the end product is satisfactory. More effective methods and techniques are urgently needed.

Some difficulties we have had on the *Nautilus* and the *Seawolf* will indicate what I have in mind. The *Nautilus* heat exchangers have tube sheets several feet in diameter and about one foot thick; they are made from stainless steel forgings. In order to obtain four satisfactory tube sheets, eight had to be made; the first four forgings had to be rejected because of poor quality.

The pressure vessel for the *Nautilus* consists of a cylindrical shell and a hemispherical head. The shell and the head are each separately heat treated. The reactor core is then inserted, and the head is placed in position and welded to the shell. Because the core might be damaged in the process, the complete pressure vessel assembly could not be heat treated as required by code practice. As a result the weld was backed up with large bolts capable of taking the entire pressure load. This added complexity, weight, cost, and delay to the job.

We had no evidence that the weld alone was not adequate. It was simply the case that no technical information existed to tell us whether it was or was not. Since we couldn't take a chance, and there wasn't time to find out, we designed the pressure vessel to use bolts, in addition to the weld. In other words, an expensive complication had to be introduced because adequate knowledge was not available.

The *Seawolf* has also had its problems with stainless steel forgings for heat exchanger tube sheets. Carbon steel nuts and bolts were found embedded in the stainless steel forgings. These inclusions were discovered after machining of the tube sheets had commenced. Apparently they remained unmelted in the original ingot, and were not detected during the forging operation. This certainly indicates that quality control can be improved.

Currently we are experiencing a series of difficulties with conventional materials. Unbonded areas have been found in stainless steel roll-bonded to carbon steel. In another case, a large carbon steel forging for a pressure vessel exhibited internal cracks; these were detected after considerable work had been done on the forging, and were found to be caused by shrinkage voids.

Many stainless steel castings for valve bodies have been porous—in fact, it is very difficult to obtain sound castings consistently. We have become reconciled to having extra parts made in order to be certain of obtaining sufficient good ones. And we still have not learned why the difficulties arise and how sound parts can be assured.

One of the tools we use for quality control is "ultrasonics." This is a potentially useful method for non-destructive testing. However, we do not feel that we can safely rely on it because there is still no accepted standard to tell us what ultrasonic indications actually mean.

Now, let me give you an example of what can happen when material is not properly identified. Shortly before the completion of the *Nautilus* we discovered that some welded pipe had been installed in the steam plant instead of seamless pipe which had been specified. Removal and replacement of the wrong piping caused months of delay in the completion of the ship and considerable extra cost. A storekeeper had issued the wrong kind of pipe. The specifications had permitted the pipe to be marked inadequately. As a result of this an opportunity existed for a mix up in the pipe racks, and a mix up eventually occurred.

The responsibility for this failure must partly be borne by the producers of the pipe and by those who wrote the specifications. Analysis of the specifications showed that they were difficult to interpret. Investi-

gation showed that similar instances had occurred in other naval vessels, and it is possible that this has also occurred ashore. As a result of this instance the Government stepped in and issued an additional military specification which requires all ferrous pipe purchased by the Department of Defense to be plainly and continuously marked. Also, the Atomic Energy Commission has adopted this military specification.

You may say that the Government was responsible in not having insisted on proper marking. I will agree. But after all there was in existence a society for the testing of materials and we had depended on them. Perhaps there should be more active consumer representation on the society's committees.

But what of non-ferrous piping and of plates and shapes? The advances in technology in recent years have forced the use of material at levels of temperature, pressure, tensile strength and corrosion resistance much nearer their upper limits than has heretofore been the case. And this trend is continuing. It may interest you to know that the Atomic Energy Commission and the Department of Defense are presently working to obtain a specification for continuous markings of non-ferrous piping and of plates, shapes and bars, both ferrous and non-ferrous. I am sure that you will cooperate in every way so we can eliminate this source of difficulty and danger both for military and industrial uses.

We in government are very hesitant to enter into fields of this type. We believe it is properly the function of industry to take care of its own products. I hope that from now on you will take the initiative in matters of this sort, so that the products on which you have expended so much care during manufacture are not improperly used.

I am sure that by this time some of you will have recalled the story of the farmer who hired a mule tamer to tame his mules. The farmer was aghast when the mule tamer began by taking a full swing with a sledge hammer on the mule's head.

"I hired you to tame the mules, not kill them," said the farmer.

"Sure," said the mule tamer, "but you have to attract their attention first."

Our attention must be attracted first to facts. After we have corraled them we can try to tame them with scientific and engineering effort.

I have presented some of the stubborn facts which hinder the development of atomic energy. Permit me to make some observations on how the situation may be improved.

Metallurgy has reached a crossroads in its history. It needs to convert itself from an art into a science. The problems presented by nuclear energy merely serve to focus attention on the urgency of this need.

Dr. Cyril Smith, Director of the Institute for the Study of Metals, at the University of Chicago is a leader in the program for transforming metallurgy into a science. He has expressed his thesis in an article which was published in the January 1952 issue of *Metal Progress*.

I will quote briefly from Dr. Smith's article.

> The metallurgist from the earliest days when he first produced the bronze and the iron tools whose importance was sufficient to name whole eras of civilization, has been purely practical. The justification of his work has lain solely in its material products, and not in what he thought about them . . . The very basis of his success lay in his disregard of theory and his willingness to try empirical experiments. But there is a limit to what a man can do in a lifetime, and empirical knowledge . . . cannot easily be transmitted to grow from generation to generation in the manner of a body of organized knowledge tied to a conceptual framework. . . . The empirical stages of knowledge are generally developed by people with short-range motivation and . . . uncultural curiosity who collect data in isolated fields without regard to their connection with others. . . . The scientist, however, seeks underlying laws applicable to all physical phenomena, and eventually it will be he who, by showing the relations of any given observation with all others, makes previously complex relations simple. . . . In this century, for the first time in history, it is possible to attempt a description of the properties and behavior of metals in terms of the same concepts that apply to all other matter. . . . The more fundamental the research . . . the more certain it is to be of eventual use somewhere, but the less obvious its immediate utility. . . . In the long run, fundamental scientific research will discover important new principles which directly open up profitable technological areas . . . It is currently impossible to predict with certainty what kind of alloy will result on alloying any two or more metallic elements, but the search for the principles is progressing rapidly, and in a few decades it will probably be possible to design an alloy with the certainty that one now designs a machine for a given purpose . . . The dollar value for such knowledge will be enormous.

Dr. Smith's clear statement of the case can hardly be improved upon.

As I have stated, nuclear energy has brought into sharp focus the necessity for building metallurgy on a scientific basis. But it has also provided powerful tools which can help to achieve this objective. One example is radioactive tracers. Another is radiation damage studies which may also become a means for understanding the physical metallurgy of metals. For example, a problem of long standing is the phenomenon of the brittle transition temperature. Below this temperature fractures are brittle; above this temperature they are ductile. It has been found that exposure to nuclear irradiation raises the brittle transition temperature. This may furnish a valuable tool for solving what is presently a mystery.

Today, metallurgists need no longer work alone. They can become part of a closely knit team which embraces many disciplines. Physicists and chemists especially are helping to get a better understanding of metallurgy because the problems are now partly in their realm. It may be that nuclear energy will be the catalyst for wiping out the fading boundary between metallurgy, chemistry and physics. After all atoms do not obey different laws for metallurgists than they do for chemists or for physicists.

It should be obvious from what I have said that many difficult problems lie before you. Metallurgy holds the key not only for the rapid development of atomic power but for many other fields as well.

But with this tremendous task in sight, it is discouraging to note that according to figures prepared

by the Advisory Committee on Metallurgical Education of your Society, the number of engineering graduates in the United States and Canada with Bachelor's degrees in your field dropped from 476 in 1950 to 204 in 1953. By 1955 the number had increased to 486. However from 1950 to 1955 the total enrollment of undergraduates and graduates steadily dropped from 3887 to 3050.

The satisfactory resolution of this situation will require a sustained and coordinated industry-wide attack over a period of many years. The efforts of your Society toward this end are very commendable. But, beyond this, it may require the establishment of special schools by your industry, and a permanent system of recruitment and support of students. It may also require a very liberal policy of university instruction for your present employees, both at undergraduate and at graduate levels.

With the existing shortage of scientists and engineers in the United States, and the rapid development of all technology, it is certain that for many years to come the shortage of trained personnel will not be satisfied.

An essential function of management and leadership, both in government and in industry, is to plan for the future. What we do today was largely determined for us by the vision and action of those who preceded us. Likewise, tomorrow's events will depend on what we plan and do today—on the wisdom we use in planning for the future.

You are faced with the tremendous problem of converting metallurgy from an art to a science. To do so will require the efforts of a very large number of skilled scientists and engineers. The major problem which you face is the recruitment and training of these people. In my opinion those organizations which take heroic steps to solve this problem will be tomorrow's leaders.

More than 20 centuries ago Archimedes said, "Give me a fulcrum and I can move the world." If he were alive today he might say, "Give me a suitable metal and I can contain temperatures as high as those of the sun."

TECHNICAL PROGRESS IN MARINE ENGINEERING DURING 1955.

ACKNOWLEDGEMENT

This staff article was published in the January, 1956, edition of "The Shipbuilder and Marine Engine-Builder."

INSTALLATION, DESIGN AND OPERATION

AN EXCHANGE of ideas between designer and operator is a healthy sign, and one which is bound to lead to an improvement in both design and operation. In the shipping industry of the United States of America, such an exchange is being directly fostered by a fourth research group, recently set up by the American Society of Naval Architects and Marine Engineers —the Ship Technical Operations Committee—one of whose principal aims is the narrowing of the gap between the designers, engine-builders, shipbuilders and "the men who go down to the sea in ships." In pursuance of this objective, a symposium on "Ship Operation" was presented at the annual meeting of the Society in November, and it will be found to be most instructive and stimulating by all who are interested in ships. Part I.—"How They Perform"—written by Mr. R. O. Patterson, captain of the S.S. *American Manufacturer;* Part II.—"Some Aspects of Merchant-ship Layout from the Marine Engineer's Viewpoint"—was contributed by Mr. Carl H. Rehbein, chief engineer of the S.S. *Mormacport;* Part III. —"Steamship Operation"—was presented by Mr. F. I. Owen and Mr. R. E. Mackey, of a shipbuilding and repairing yard; Part IV.—"An Operator's Comments on the Air-conditioning of Ships"—was by Mr. John E. Bone, assistant port engineer of the American Export Lines; Part V.—"The Relationship between Naval Architecture and Vessel Maintenance Costs" —was by Mr. John M. Van Orden; and Part VI.— "The Loading and Unloading of Tankers"—was written by Mr. F. M. Hillman.

Such a wide field is covered that a short *précis* would be misleading, and it is not attempted here, but the symposium, when published with its final discussions, will repay study.

On the particular subject of correlation of model data and ship records, a second most valuable paper was presented in April last, as a joint paper read by Professor G. Aertssen, before the Institution of Naval Architects and the Institute of Marine Engineers, and entitled "Sea Trials on a 9,500-ton Deadweight Motor Cargo Liner." The vessel whose trial results are reported is the cargo motorship *Lubumbashi.* Records have been collected and are presented for ship speed, power, fuel consumption, propeller thrust and revolutions, ship movement, wind and waves under varying conditions of draft, weather and fouling. The method of data analysis is given, and an estimate is made of the efficiency and economy of the ship and her machinery in different service conditions.

A large part of the test program is devoted to ship-model correlation, and, to this end, two measured-mile trials were carried out, one in ballasted condition, and a second in a loaded condition at the commencement of the maiden voyage. A model has been run at the Ship Division of the National Physical Laboratory, and it is found that, by using the N.P.L. method, model predictions agreed with the ballasted trial for a ship correlation factor of 1.0 (for flush-welded butts and riveted seams), while for the loaded trials, a correlation factor of 0.95 appears to be more appropriate.

The important economic factors influencing choice of machinery for a modern cargo liner are reviewed in the 27th Thomas Lowe Gray Lecture, entitled "Some Factors in the Selection of Machinery for Cargo Liners," given by Commander (E.) L. Baker, D.S.C., R.N. (Rtd.), before the Institution of Mechanical Engineers, in January, 1955. The author particularly considered the effect of the specific fuel consumption of various types of propulsion machinery; the effect of the type of machinery on Admiralty coefficient; the effect on the choice of machinery; the effect of various propulsion machinery installations on voyage costs; and the tendency towards machinery standardization in new ships for the fleet of the Blue

Funnel Line. Data are presented in both tabular and graphical form, and an example is given, showing how the shipowner's requirements are co-ordinated with hull designs and machinery requirements, in order to arrive at a final complete specification for both hull and machinery.

A symposium was published in May last, jointly by the Institute of Marine Engineers and Institution of Naval Architects under the title of "Advanced Machinery Installations Designed for the Maximum Saving in Weight and Space." In order that there should be a basis for comparison between the various proposals submitted in the six papers comprising the symposium, the authors were asked to prepare designs for the machinery for two existing ships (of which the plans were reproduced in the symposium), also the essential particulars regarding weight and space. The smaller vessel (length B.P. 400 ft. 0in.; 5,089 tons gross) requires 3,340 S.H.P. for 12½ knots, while the larger vessel (length B.P. 470 ft. 0in.; 9,718 tons gross) requires 10,500 S.H.P. for 16½ knots. The papers submitted were as follows:—

"Gas-turbine Engined Installations," by Mr. T. W. F. Brown, D.Sc., S.M., contains full details of engine-room auxiliaries, complete with weights, sizes and estimated fuel consumption. The maximum cycle temperature is 1,250 deg. F. and astern power is provided by means of an hydraulic torque converter.

"Proposals with 10,500-S.H.P. Sulzer Diesel Engines in a 10,600-ton Cargo Ship," by Mr. E. W. Cranston, Wh.Sc., D.I.C., covers *(a)* directly-coupled eight-cylinder supercharged engines at 115 r.p.m.; *(b)* two eight-cylinder supercharged engines at 240 r.p.m., coupled together and to the propeller shaft through hydraulic reduction gear (propeller shaft at 100 r.p.m.); and *(c)* two Vee twelve-cylinder supercharged engines at 420 r.p.m., coupled together and to the propeller shaft at 100 r.p.m., through hydraulic reduction gear.

"Direct-drive Diesel Machinery (Doxford)," by Mr. P. Jackson, M.Sc., covers three proposals for each ship.

"Steam-turbine Machinery," by Mr. E. H. Smith, suggests the use of 650 lb. per sq. in. (gauge) at 950 deg. F. for initial steam condition for the large power, and 600 lb. per sq. in. (gauge) at 800 deg. F. for the lower power.

"The Napier Deltic Marine Engine," by Mr. E. E. Chatterton, B.Sc. (Eng.), F.R.Ae.S., and Mr. D. D. Fuller. The engine speed proposed is 1,500 r.p.m., the drive being taken through a reverse-reduction gear-box containing two constant-mesh systems (one for ahead and the other for astern running). The appropriate gear train is connected by hydraulically-loaded plate clutches. Six standard engines are coupled together for the larger vessel and two for the smaller.

"Geared Gas-generator/Turbine Machinery with Free-piston Gas Generators," by Mr. D. D. Williamson. The smaller installation is provided with four free-piston gas generators, supplying a reversible gas turbine, which is coupled through mechanical reduction gearing to the propeller shafting. For the larger vessel, 12 free-piston gas generators are suggested, supplying two reversible turbines, both being coupled together and to the propeller shaft through mechanical reduction gearing. An "engines aft" installation is also put forward for the free-piston proposals.

The successful operation of a fleet of ships requires as great care in the planned maintenance of them, as in the initial design. This aspect of ship operation, in so far as it affects the Royal Navy, is dealt with in a paper, entitled "Changing Pattern of Maintenance and Repair of the Machinery of the Fleet," which was read by Rear-Admiral John E. Cooke, R.N., before the Institution of Mechanical Engineers, at Southampton, in July last. The author indicated the changes which have taken place in the maintenance and repair of the machinery of Her Majesty's ships during the past 40 years, the reasons leading to the changes and future trends. Convincing evidence is given of the value of preventive maintenance, in which there is a planned routine of servicing, examination and corrective action, before excessive wear or breakdown occurs.

In particular, mention is made of the effectiveness of chemical descaling of the sea-water side of heat-exchangers and of electrolytic derusting. For chemical descaling, a proprietary mixture of hydrochloric and phosphoric acids, together with inhibitor, is used, care being taken to protect any ferrous parts with a rubber-based paint. The results have been most satisfactory, there have been no cases of tube or tube-plate attack, and in the case of badly scaled main condensers, the result has been to enable the vacuum to be restored to the initial design figure. Electrolytic derusting has been successfully used for cleaning turbine blades and rotors *in situ*, and has also been developed for the removal of rust from decks. External boiler-cleaning by means of water washing is found to be efficient and a great time saver, the refractory brickwork, of course, being protected with a bitumen-based paint. The initial lighting-up after cleaning serves to burn-off the paint, and leaves everything ready for use.

INSTRUMENTATION

The necessity for the proper instrumentation of marine power plants is now widely accepted, but the same cannot be said of automatic and distance control. Nevertheless, the necessity for, and advantages of, such controls increase with the value and complexity of an installation. It provides a means whereby fewer men are required to control greater power, and it thus offers a possible solution to the existing shortage of engineers, a shortage which is likely to persist, and even increase, with the increased demand for technically-trained men ashore. Automatic and distance control also provides greater safeguard for valuable and complex plant, than does manual operation. The probable future development of gas-turbine machinery and of nuclear steam-raising plant will require a considerable use of automatic and distance controls,

if such plant is to be properly safeguarded and operated at peak efficiency.

In a paper entitled "The Application of Modern Instrumentation and Distance Control of Machinery," by Mr. R. H. Paddon Row, which was presented to the Institute of Marine Engineers, in March last, the general principles, together with particular examples of instrumentation and distance controls, as applied to marine mechanical (as opposed to electrical) machinery are discussed. Various types of instruments used for measuring and indicating pressure, temperature, flow rates and other such usual quantities, together with their grouping and location, are dealt with. This latter is of utmost importance, if the maximum benefit is to be obtained from instrumentation. Distance and automatic control instruments are similarly treated, and the scope of such equipment is discussed, with particular reference to the operating medium and suitability for use in ships. The problems of control at sea are mentioned, and are compared with established practice in land installations. The necessity for regular maintenance and adjustment of instruments and controls is emphasized. As the scope of instrumentation increases, it is recognized that a specialist is necessary for proper maintenance, although, at the present time, it is possible for an engineer to learn enough in a three months' course to enable him to carry out day-to-day maintenance on instruments fitted on main propulsion machinery.

The paper entitled "Instrumentation in the Marine Industry," by Mr. Frank Newman, which was read before the American Society of Naval Architects and Marine Engineers, in November last, poses the question of how is the American marine industry to cope successfully with low-cost competition, and suggests that one course is to place its faith in the traditional tactics of introducing technological advances to bring about lower operating costs. The value of automatic control is analyzed, and examples are given of instrumentation to assist in the reduction of fuel consumption and maintenance. Potential labor savings are explored, and the principles of operation of some modern instruments are described and illustrated.

NUCLEAR POWER

A survey of British and American unclassified material relating to the use of nuclear power for marine propulsion is made by Mr. K. Maddocks, B.Sc. (Tech.), in his paper entitled "Nuclear Power for Commercial Vessels," which was read before the Institute of Marine Engineers, in December last. It makes a worthy sequel to the paper entitled "Atomic Propulsion, with special reference to Marine Propulsion," by Sir John Cockcroft, K.C.B., C.B.E. F.R.S., which appeared in the Transactions of the Institute of Marine Engineers, for April, 1953. Mr. Maddocks' paper opens with a brief discussion on the principles of fission and reactor operation. It is at once clear that the control room will be completely isolated from the reactor space, so that all the factors affecting operation must be metered on distant recording instruments in the control room, while much of the plant operation will be subject to automatic controls. In particular, possible radiation hazards in the form of gamma rays or escaping neutrons must be detected by an elaborate system of instruments located both inside and outside the reactor. A marine installation will also require these instruments on the ship's hull, and on the ventilating and sanitary systems.

An important part of the paper deals with five types of reactor, which could be suitable for marine use, and also with one type for fuel production. (This latter type is unsuitable for use in ships, due to the extensive ancillary chemical plant and shielded material-handling equipment to be accommodated.)

A pressurized water reactor, as fitted in the U.S.S. *Nautilus* has become the pioneer reactor. Highly purified water, under pressure, forms both reactor coolant and moderator to control the reactor operation at a comparatively low level (thermal operation). Heat is transferred from the primary circuit to the secondary circuit in a heat-exchanger/steam generator, the secondary circuit being probably forced circulated. An obvious disadvantage of such a system is the impracticability of generating steam with any appreciable superheat, as, even if the pressurized coolant is at 1,000 lb. per sq. in., its highest temperature must be below the saturation temperature of 544 deg. F., which limits the maximum temperature to (say) 534 deg. F., equivalent to 60 deg. superheat at 400 lb. per sq. in. Although not an economical source of power, it is the type on which most experience is available.

The boiling-water reactor is one in which steam is generated by the primary water circulating the core, thus cutting out the need for a heat-exchanger and secondary circuit. The simplification, however, is produced at the cost of introducing a secondary shielding problem, due to the radio-activity of the steam-turbine-condenser system, and also a variation in density of neutron leakage due to the boiling of coolant and moderator, thus causing a fluctuation in power level of the reactor. Again, only low level of operation is possible, and only saturated steam can be generated.

Some of the inherent limitations of the heterogeneous reactor (of which the two mentioned in the foregoing are examples) can be overcome by using a homogeneous reactor operating on an intimate mixture of fuel and coolant/moderator in the form of a solution of uranium salt in ordinary water, which, itself, forms the primary circuit. Steam generation is by heat exchange, so that the temperature ceiling is again restricted to about 530 deg. F. One of the most striking features of the design is the absence of control rods, the system having proved itself to be self-regulating. Shutting down the circulating pump cools down the heat-exchanger, with resulting increase of density of the uranium-salt solution outside reactor core. Meanwhile, within the core, on account of the cessation of circulation, the density decreases until control is established, due to the increased neutron leakage thus allowed. On recommencement of circulation, the introduction of more dense solution into

the core pushes up the output of the reactor until the design level is re-established.

Liquid sodium has a boiling point of 1,600 deg. F., and can be used as a coolant, thus permitting the production of moderately superheated steam at, say, 800 deg. F. level. Sodium, however, is a poor moderating medium, and, thus, a separate moderating system is required, which could be of the well-established graphite-block construction. A version of this type is to be used in the U.S.S. *Sea Wolf*. There is a considerably increased shielding problem, and the danger of sodium leakage is twofold, owing to its high degree of radio-activity, and its violent reaction with water. A typical specification for leakage tolerances (including pump glands, if any) is one cubic centimeter in 10 years.

The gas-cooled reactor is at present Britain's choice of development as a power reactor ashore. The reactor coolant gas (in a closed circuit) circulates a heat-exchanger/steam generator as in waste-heat boiler practice, generating steam at moderate superheat up to a temperature of, say, 750 deg. F. Moderation has to be by block graphite, and control is exercised by the positioning of rods of either boron steel or cadmium.

Preliminary designs and outlines of equipment, which have been prepared for a submarine installation to compare the use of water, sodium and helium as coolants, appear to favor helium, although, of course, confirmation must await until considerable further development occurs.

The latter portion of this paper is devoted to the discussion of coupling together a gas-cooled reactor and closed-cycle gas turbine, which combination is considered by the author to offer the most favorable balance of first and operating cost, and one which offers simplicity and safety in operation. The cycle proposed using helium as reactor coolant, and also helium in the closed-cycle gas turbine, is outlined in reasonable detail; the maximum temperature is taken at 1,500 deg. F.

When discussing the economic aspect of the use of nuclear fuel, it is pointed out that, as the amount of shielding required is practically independent of power output, nuclear-power plant of low output will be excessively penalized, and it is considered that, at the present stage of development, only plants of 10,000 S.H.P. and above could be favorably considered. The bigger the power, the greater the advantage favoring nuclear energy generation. In addition to maximum power output, a further factor of prime importance is the load factor, determined by the percentage of time spent under full power. For general cargo-carriers, this load factor may be as low as 26.6 per cent; such a low figure considerably reduces the suitability of nuclear propulsion. A survey of desirable conditions for the successful operation of nuclear-powered ships quickly points to highly powered ships with high load factor, preferably operating between terminal ports possessing facilities for handling radio-active material. Bulk carriers and oil-tank ships are thus indicated, a further point in favor of the latter being the accomplished fact of offshore discharging and loading of oil cargoes. Such a measure of isolation provides an added measure of safety.

In connection with nuclear-energy generation, a very useful series of articles appeared in the issue of *Power Works Engineering*, for January, 1955, in which the basic principles and terminology used are explained. A brief account of the fundamental concepts of nuclear fission is given, together with an outline of the main processes and instruments of this new industry.

GAS TURBINES

A naval propulsion plant for medium and large vessels usually consists of a cruising turbine and a full-power set. A similar combination of gas-turbine machinery was investigated by Mr. D. F. Collins, B.Sc. (Eng.), and Mr. D. W. Thomas, and was reported in their lecture to the North-East Coast Institution of Engineers and Shipbuilders, in March last, under the title of "A Comparison of the Total Weight and Bulk of In-line Naval Gas-turbine Engines of Differing Degrees of Complexity." A cruising turbine set, supplying up to 33 per cent of the maximum required S.H.P., is permanently coupled to the propeller; while the remaining power is generated by a separate turbine set, which is coupled on to the shaft when required. It is suitable, therefore, for the cruising set to be of a complex high-efficiency type with a low fuel consumption, while the main power turbines (being only occasionally used) can be of a simpler design, as fuel economy at full power is less important than under cruising conditions. The weights and sizes of four basic power plants were estimated from detailed design studies and an operating schedule was taken which was typical for medium-sized naval vessels.

The authors conclude that, of the plants considered, the one using two double-compound intercooled engines, with heat exchange on the cruising engine, yields the lowest value of total weight of plant and fuel, and of size of deck openings. The plant, consisting of two single-compressor engines with heat exchange on the cruising engine, gives the shortest machinery length, though this plant is among the worst as regards overall bulk and size of deck openings. The other machinery installations that are discussed are a plant consisting of two intercooled double-compound engines and double-compound plant which employs neither intercooling nor heat exchange.

The near future is likely to see at least two big all-gas turbine ships afloat, both being conversions. The motor vessel *Auris* is being stripped of her present machinery, comprising three Diesel-generator sets and the famous pioneer marine gas-turbine set, and is being re-engined with a single 5,500-S.H.P. gas-turbine unit, coupled to the propeller through mechanical gearing. The experimental character of the vessel is thus being retained.

The conversion of a "Liberty" ship, the *John Serg-*

eant, is being proceeded with in the United States of America, the gas-turbine set being designed for a maximum continuous rating of 6,600 S.H.P. An open-cycle, two-shaft regenerative machine is being installed, incorporating a 14-stage axial-flow compressor, a combustion system, a single-stage high-pressure turbine driving the compressor, and a single-stage low-pressure turbine driving the propeller through reduction gearing. Full details of the proposed installation were presented by Mr. J. J. McMullen in a paper, entitled "Gas-turbine Installation in the 'Liberty' Ship *John Sergeant,*" which was read before the American Society of Naval Architects and Marine Engineers, in November last.

In January, Mr. I. G. Bowen, B.Sc., and Mr. W. Tipler, M.A., read a paper entitled "The Choice between Single and Multi-combustion Systems for Gas Turbines," before the North-East Coast Institution of Engineers and Shipbuilders. They point out that the diversity of the applications for which gas turbines are being considered is matched by the wide variety of their external configurations, even when limited to the combustion of fuel oils. This paper examines the influence of the type of combustion system selected on the overall design and layout of the engine, and the influence of the combustion system selected on the design of the individual combustion chambers comprising the combustion system. The principal conclusion is that, in spite of its greater bulk, the single-chamber combustion system is to be preferred, on the grounds of reliability and ease of maintenance.

For satisfactory burning of residual fuel in open-cycle gas turbines, great care and attention must be paid to the preparatory treatment, and, in this connection, a paper entitled "Modified Residual Fuel for Gas Turbines," by Mr. B. D. Buckland and Mr. D. G. Sanders, was read at the 1954 annual meeting of the American Society of Mechanical Engineers, held in New York, in November, 1954. This paper described a method of treating residual fuels, which has been very effective in reducing oil-ash corrosion of blading and burner nozzles. In locomotive service, units have run for periods of up to 100,000 hours (consuming about 170,000 tons of treated oil), and three power-generating units have burned more than 20,000 tons, with no apparent corrosion. The treatment consists of washing out more than 90 per cent of the sodium salts in the fuel, followed by the addition of a water solution of magnesium sulphate just before the fuel is burned. A specification is proposed, defining a fuel which can be obtained at the point of use by means of this de-salting method.

STEAM TURBINES AND BOILERS

Steam-turbine installations continue to be the natural choice for large-power single-screw ships, and while many notable vessels have used steam conditions of 600 lb. per sq. in. at 900 deg. F., most owners are content to have initial conditions in the region of 450 lb. per sq. in. at 800 deg. F. The advancement of evaporating capacities and of stop-valve conditions (in particular, the superheat temperature) raises many problems in steam-raising and in piping materials and connections.

In connection with steam-raising, the results of an "Investigation into the Density Distribution, Water and Steam Velocity, and Pressure Loss in Vertical and Horizontal Boiler Tubes," has been published by Mr. K. Schwarz in Research Report No. 445 of Verein Deutscher Ingenieure. Much of the published literature is discussed, and the results of original experiments on a full-sized experimental boiler are reported. As the title indicates, records of mass flow, distribution of density, and water and steam velocities in both vertical and horizontal tubes have been made. The derived friction coefficients are considerably different from those usually accepted for water-steam mixtures. Such a difference has a marked effect on the estimated condition of circulation, in natural-circulation boilers especially.

The problems of welding steam pipes, suitable for working between 850 deg. F. and 1,100 deg. F., are discussed by Mr. J. G. M. Turnbull, B.Sc., A.R.I.C., F.I.M., in a paper entitled "Welding of High-pressure Steam Pipes," which appeared in the issue of the *British Welding Journal,* for January, 1955. After commenting on the various materials used for the manufacture of steam piping for such conditions, the author discusses the welding and heat-treatment required to produce satisfactory joints, and also reviews the non-destructive methods of testing, special reference being made to gamma radiographic technique.

The Technical Information Report No. 42 of the National Research Council gives a very extensive bibliography on radio-isotopes and their uses in engineering. Of particular interest is the use of a radio-active capsule to aid weld inspection. For the examination of a circumferential joint, the radio-active source is placed at the center of the section, and a strip of film is put in position around the joint. The developed film gives a complete record of the entire welded joint and can be taken with one exposure.

In connection with the actual welding of pipes, a method of providing an inert-gas backing is described by Mr. M. Lorant in the March issue of *Sheet Metal Industry.* The method, devised by Mr. R. T. Pursell, makes use of paper discs to contain the inert gas in the locality of the weld, and tests using argon backing have given consistently good results. Copper, aluminum and austenitic stainless steel have all been satisfactorily welded. It is concluded that the use of an inert-gas shielded arc-welding process for the welding of pipes (together with inert-gas backing) leads to an improved root bead condition and at a favorable cost, when compared with the use of a backing ring.

In searching for an explanation of the variability of creep-strength exhibited by carbon steels, Mr. W. E. Bradgett and Mr. M. G. Gemmill conducted creep and other allied tests on samples and specimens taken from 58 different basic open-hearth casts of carbon steel. The results of their work are reported in the March issue of the Journal of the Iron and Steel Institute. One of the factors examined is the aluminum addition, which is varied from 1 to 2½ lb. per ton. This

variation is shown to be very significant, and it is found that the soluble aluminum present in steel is a paramount factor in determining creep behavior; the McQuaid-Ehn grain size is also related to it. The view is presented that the creep resistance of the steels tested correlates directly with the aluminum-nitride content, which, in turn, is related to the soluble aluminum-nitride grain-size characteristics of the steel. From the practical point of view, it would appear that steels of good creep resistance can be obtained with aluminum additions of up to 1½ lb. per ton, together with suitable limitation of the mold addition of aluminum.

The British Electrical and Allied Industries Research Association have reported on the cracking, in service, of 0.5 per cent molybdenum-steel piping. It appears that the incidence of failures is largely associated with two-shift operation, and its accompanying heating and cooling cycle each day. Although laboratory experiment reveals no substantial difference in the rupture strength, at service temperatures, between 0.5 per cent molybdenum steel and 1.0 per cent chromium, and 0.5 per cent molybdenum steel, the latter has a superior ductility, with markedly greater elongation at rupture. In the light of experience, it is recommended that, where frequent heating and cooling are to be encountered in operation, the chromium molybdenum steel should be chosen in preference to the 0.5 per cent molybdenum steel.

DIESEL MACHINERY

The past year has seen the first Doxford diaphragm engine and the first Doxford supercharged engine running at sea, both with very satisfactory results. The diaphragm engine has six cylinders 750 mm. bore, a combined stroke of 2,500 mm. and a normal rating of 8,850 S.H.P. at 110 r.p.m. Included in the usual tests was a 72-hour non-stop run on fuel oil having a viscosity of 1,500 seconds Redwood No. 1 at 100 deg. F. At the end of this period, the amount of sludge collected in the diaphragm chambers was only a few pints. It thus serves the double purpose of reducing the quantity of lubricating oil passing from the crank chamber to the cylinders, and also of the carbonized oil from the cylinders passing to the crank chamber.

The first large supercharged engine has also been delivered by Messrs. Harland & Wolff, Ltd. This engine has six cylinders, 750 mm. bore, a combined stroke of 2,000 mm., and a service rating of 8,000 S.H.P. at 112 r.p.m. The engine is similar in design to the non-supercharged engine; but the usual Roots scavenge-air blower is replaced by Napier turbo-blowers, while Wilson and Kyle gas-compression fuel pumps and fuel valves are used. The turbo-blowers each consist of a single-stage axial-flow turbine and a centrifugal air compressor, mounted on a common shaft; each of the two blowers serves three cylinders, being mounted directly on the back of the cylinders and making use of the exhaust blow-down energy. Pressure-charging increases the continuous non-supercharged rating by 30 to 35 per cent.

In a different range of size, Ansaldo S.A., Genoa, have fitted supercharged engines in the fishing vessel *Nuovo Stella*, of 196 tons displacement. The claims for a supercharged engine in small fishing vessels are:— (1) A 40 per cent reduction in weight, (2) a 40 per cent reduction in overall length, and (3) a reduction in fuel consumption of 5 per cent at full power and of up to 15 per cent at part load. It is claimed that the rate of supercharging, which may be 50 per cent when running free, can be raised to as much as 80 per cent when towing, without overloading either the engine or the supercharger.

Of considerable interest to Diesel designers is the paper entitled "An Investigation of the Factors Contributing to the Failure of Diesel-engine Pistons and Cylinder Tops," by Mr. D. Fitzgeorge, B.Sc. (Eng.), Ph.D., and Mr. J. A. Pope, Ph.D., D.Sc., Wh.Sc., which was read before the North-East Coast Institution of Engineers and Shipbuilders, in February last. The authors discuss the stresses occuring in cylinder heads and pistons. Particular attention is given to the nature of the stresses, their magnitude and the likelihood of eventual failure. The physical and mechanical constants affecting thermal stresses are noted, together with the modifying influence of plastic deformation. It is shown that cracks commencing at the hot side of a piston are initiated by high-tensile stresses set up when cooling down, and these, in turn, are due to the plastic deformation (in compression) which takes place at the hot running condition. Cast-iron, with its low ductility and strength in tension, would be most liable to failure from such causes, and the authors claim they have not encountered such failure in steel components. Any hot spot in cylinder cover or piston is a potential source of danger, particularly if it is aggravated by thermal-stress raisers, due to assymetry in design.

A second main cause of failure is the fatiguing action of the repeated combustion loads combining with thermal stresses due to normal temperature gradients in the walls. The resulting cracks occur at the horizontal (circumferential) junction of the crown and wall of pistons, and at points of high thermal stress in cylinder heads, and are initiated on the cooled side of the member, occurring in both steel and cast-iron components.

Typical failures are discussed and guidance charts are given, aimed at assisting designers to overcome faults which show up in particular designs. Alternative designs are suggested, which could be effective in overcoming cracking troubles in the pistons of highly-rated engines.

A source of failure that has received considerable attention in recent years is the crankcase explosion, and a paper entitled "Crankcase Explosions in Marine Engines," by Mr. J. H. Burgoyne, D.Sc., Ph.D., F.R.I.C., and Professor D. M. Newitt, M.C., D.Sc., F.R.S., which appeared in the August Transactions of the Institute of Marine Engineers, indicates that most of such explosions are likely to be associated with the formation and subsequent ignition of a lubricating-oil mist, in contact with an overheated engine part. Observation shows that, even at the lower explosive

limit, the optical density of an oil mist is very high, and at the upper explosive limit, is even higher; whereas in an engine under normal conditions, the lubricating-oil mist is barely visible. The development of a warning system, based on the interruption of a beam of light, is proceeding and shows great promise.

Methods of limiting the effects of an explosion are studied, and it is concluded that the provision of relief ports can be very effective, provided attention is given to detail. The area of relief should be as large as possible, and, preferably, should be provided in the form of a smoothly operating spring-loaded plate valve opening at explosion pressure. Particular attention must be given to the mode of retarding the plate or other moving part at the end of its travel, so that damage may be avoided. There must also be flame traps to protect engine-room personnel from danger of issuing flame.

The attack on the problem of reducing cylinder wear when using heavy fuel is twofold, one by development of special wear-resisting lubricating oils and the second by careful use of centrifugal cleaning plant. In the first line of attack, the Shell Company have developed a new cylinder lubricant (Shell oil S.4264) having wear-reducing properties, which is intended for use in engines using heavy fuel. The oil is in the S.A.E.50 class, and contains additives to prevent corrosive wear. Comparative tests have been conducted with different types of engines, both afloat and on the test-bed, using a normal oil in some of the cylinders for purposes of immediate comparison. The effect of S.4264 oil in reducing wear rates is marked. The following figures are from a 1,500-hour test on a Burmeister & Wain double-acting, two-stroke cycle engine, 590 mm. bore and 1,700 mm. stroke, and of 7,000 S.H.P.:—Maximum wear rate of cylinders using S.4264 oil is 0.12 mm. per 1,000 hours, for the cylinders using a straight S.A.E.40 oil, it is 0.60 mm. per 1,000 hours. The oil is non-toxic and there is no danger to health in handling it.

In the second direction of attack, tests (reported in the *Sulzer Technical Review)* have been carried out by Mr. A. Brunner to determine the best temperature and procedure for centrifuging. Tests showed that most of the foreign bodies of high specific gravity were caught in the purifier throat, and particles found on the wall of the purifier bowl were fewer and had a lower specific gravity, while, furthermore, the throat of the clarifier was usually empty. It is considered that the highest possible temperature (up to a maximum of 185 deg. F.) should be used for centrifuging, limited only by the flash-point of the oil and the vaporization of the contained water.

A theoretical analysis indicates that the action of two centrifuges in parallel should be more effective than their action in series. However, in practice, it is found that the specific gravity of the cleaning water may rise during operation, due to the dissolution of salts separated off in the process or to the admixture of salty water removed from the oil, and that such occurrences seriously impair the effectiveness of the purifier. It is thus confirmed that, in spite of the additional complexity of the installation, it is wiser to use a clarifier after the purifier and that a continuous water feed should be used in the purifier.

Concerning through-put it is concluded that it is a mistake to be sparing in the use of centrifuges, and a diagram is given showing the number of centrifuges recommended for any viscosity oil, and any particular engine output. The efficiency of centrifuges within their normal working range increases with decreasing through-put.

It is, of course, essential that any purification plant installed in a vessel should be cleaned regularly. Such cleaning is a tedious and dirty job when done by hand, and a more widespread use of the type that can be cleaned while still running (a self-cleaning type) will probably result in greater attention being paid to this matter.

STEAM RECIPROCATORS

Although the figures for the proportion of new world tonnage propelled by steam reciprocating machinery is now in single figures (approximately 5 per cent), such machinery does still persist, and in his paper entitled "Steam Reciprocating Engines," which was read before the Institute of Marine Engineers in November last, Mr. G. Yellowley gives an interesting account of the final state of design which the steam reciprocator has reached. The figures for fuel consumption are very impressive, when combined with the inherent reliability of the machinery. When working from initial conditions of 225 lb. per sq. in (gauge) and 750 deg. F. (at the boiler), the use of steam reheat gives a fuel consumption of 0.75 lb. per I.H.P.-hour. The addition of an exhaust turbine has always shown itself to be a most effective means of improving performance, and in the case of a reheated engine, the figure for fuel consumption is improved to 0.65 lb. per I.H.P.-hour for a turbine combination set.

In the case of the cargo vessels *Rostock* and *Wismar*, which have been built for the East German nationalized shipping company, the main propulsion machinery is a combination set comprising a double-compound steam engine and low-pressure turbine. Maximum power developed is 2,750 I.H.P., the service power being 2,450 I.H.P. Steam is generated in two water-tube boilers at a pressure of 240 lb. per sq. in. (gauge) and a temperature of 570 deg. F. Coal fuel is handled by a fully automatic stoking plant, including conveyors to trim it from the bunkers, breakers to reduce the size to pass a 2-in. screen, and a hoist to carry it into the hoppers which feed the grates. Ash breakers and ash hoists are also provided.

AUXILIARY EQUIPMENT

A heat pump and a refrigerating set both work on the same basic cycle. In each case, heat is abstracted from a heat source by a refrigerant, work is done on the refrigerant so that its temperature is raised, and, finally, the heat initially abstracted, together with the heat equivalent of the work put into it, is transferred to a heat sink of some kind. If the objective is to maintain a sub-atmospheric temperature at the heat

source by continually removing heat which tends to leak in, then the machinery is considered to be a refrigerator.

When interest is centered on using the heat which is discharged at the sink end for space-heating purposes, the machinery is considered to be acting as a heat pump. In such a case, the heat source is usually the sea (at a temperature of between 30 to 50 deg. F.), and the sink is a heat-exchanger (the condenser), in which is heated the fluid used for circulating the space-heating radiators, the temperature of which may well be required to be 160 deg. F. Thus, a heat pump has to provide a temperature rise of over 100 deg. F. (whereas a refrigerator seldom has to provide a temperature rise of more than 80 deg. F.). The attraction, of course, is that, in a well-designed unit, the whole of the input energy to the compressor, together with most of the heat lifted from the source, is available for space-heating.

On very large installations, the initial capital outlay can be recouped in the first few years, but many ship installations are not large enough to be economically attractive.

In an article entitled "Heat Pumps on Board Ship," which appeared in the issue of *The Shipping World*, for January, 1955, Mr. E. G. R. Roberts indicated the general conditions under which space-heating in a ship may or may not be achieved economically by heat pumps. They are: —

(1) No form of heat pump is likely to be profitable if the vessel trades largely in cold climates.

(2) No heat pump is likely to be economical unless the bulk of the plant required is already fitted for some other purpose, such as air-conditioning.

(3) Vessels in which main and auxiliary machinery are steam-driven are not likely to be suitable.

(4) Vessels with main machinery steam-driven and with Diesel generators may be suitable if not excluded by conditions (1) and (2).

(5) Vessels with Diesel-electric propulsion are most likely to be suitable, subject to conditions (1) and (2).

Using the twin-screw steam turbine-driven passenger liner *Southern Cross* as an example, it is shown that, with relatively minor alterations, the air-conditioning plant could be run, when required, as a heat pump having a capacity of 9,000,000 B.T.U. per hour, and fuel consumptions are estimated for the same heating effect produced by direct steam-heating, using main boilers, and for electrical heating, using Diesel generators as the source. Assuming an electrically-driven heat-pump compressor, analysis indicates that, when using such a pump for space-heating, a saving of 2½ tons per day is to be expected when compared with main-boiler steam-heating and 11 tons per day compared with electrical heating.

In many ships, the evaporator was a piece of equipment used only when the supply of shore-bought water ran low; the main discouragement to its regular use was the need for frequent descaling of the heating coils. This discouragement can now be removed by the use of specially developed evaporator compounds, which allow evaporators to be steamed up to 4,000 hours, without stopping for coil removal and cleaning. In Admiralty Bulletin No. 62, Mr. J. Leicester gives an account of the development of the Admiralty Evaporator Compound. One of the constituents, Belloid T.D., is a powerful amonic dispersing agent for calcium and magnesium salts, which, in an evaporating solution of brine, dissociates and induces a pronounced negative charge on the surfaces of the precipitated crystal nuclei of scale-forming material. At the same time, the anion is adsorbed on the coil heating surface, so that precipitated material does not come down as a hard, strongly adherent scale, but forms a brittle and non-adherent coating, easily broken off by thermal shock.

Two other constituents are mentioned. One is a sequestering agent, tending to collect a proportion of the calcium and magnesium ions from the brine, and also taking up any copper ions present, which may have resulted from the corrosion of the copper heating surface. The other ingredient is a foam suppressor. All constituents are said to be non-toxic.

Under the "Liberty"-ship experimental conversion program, initiated in the United States of America, the *Thomas Nelson* is being lengthened 25 ft. 0in., and is being converted to geared-Diesel propulsion. The derricks are being dismantled, and the holds are to be served with six 5-ton revolving deck cranes, running on tracks. The three forward holds are served by four cranes, using tracks laid athwartships between the hatches, while the two after holds are served by two cranes, which will be able to move fore and aft. All cranes are self-propelled, some electrically and some hydraulically, so that efficiency of operation of the two alternatives can be compared.

An increasing interest is being shown in hydraulic winches and, in this country at least, it is apparent that the sale of straight D.C. electric winches is dropping below its peak of a few years ago.

Many manufacturers are supplying A.C./D.C. winches with self-contained motor-generator units giving reliable voltage control, and straight A.C. winches have also been introduced. During the past few years, a considerable number of vessels have been built employing A.C. main supply. The advantages of such a supply are very great and increase with the size of the auxiliary load, the only big problem being the provision of a satisfactory drive to deck auxiliaries. A very useful review of the present situation is made by Mr. A. S. Brown, M.B.E., in his paper on "Review of Deck Auxiliaries Supplied to Vessels with A.C. Supply," read before the Institute of Marine Engineers, in November last. The author considered winches, capstans and windlasses and discussed the different types of drive that have been provided for them in ships having A.C. supply. By considering the whole of such deck machinery as one problem, simplifications can be introduced and the amount of spare gear required can be reduced. Even

so, such installations are usually more costly than the conventional series-resistance controlled D.C.-motor system.

VIBRATION

A paper by the late Mr. H. G. Yates, M.A., entitled "Prediction and Measurement of Vibration in Marine Geared-shaft Systems" (Transactions of the Institution of Mechanical Engineers, 1955) is a characteristically straight-forward and thorough presentation of the development of a highly successful method of predicting shaft vibrations. In the author's own words, he "felt the need for a reliable method of predicting shaft vibrations which would enable not only critical frequencies, but also vibration amplitudes, to be estimated for any given stimulus, and which, furthermore, would do so without requiring an inordinate amount of calculation work." The usual methods for calculating natural frequencies by means of Holzer tables (or some variant) is effective, though clumsy, but they cannot be readily used to provide the response to a stimulus at an arbitrary frequency. This difficulty is especially apparent when a system has two natural frequencies close together.

Mr. Yates used the concept of mechanical impedance analogous to electrical impedance, and, for any mechanical system considered, set up the equivalent electrical system. Then, making use of the well-known properties of electrical circuits carrying simple harmonically varying currents, the resulting torques and angular displacements could be estimated for any superimposed forcing torque.

The electrical impedance of an element of a circuit is the ratio between the voltage applied to it and the current through it. The three basic elements of electrical networks and their mechanical equivalents are:—

(1) Inductances equivalent to mass-movement of inertia (whose impedance is proportional to frequency).

(2) Capacitance, equivalent to shaft-flexibility (the impedance of which is inversely proportional to frequency).

(3) Resistance, equivalent to damping (the impedance is constant).

Force or torque is treated as the mechanical analogue of voltage, and the analogue of current becomes linear or torsional velocity. A simplification is introduced by using the ratio of the impedance to one fixed reference impedance, thus making them non-dimensional, and it sometimes is a further simplification to use the reciprocals of the non-dimensional impedance ratios—termed admittance ratios.

When considering the magnitude of damping forces, the author followed Mr. S. Archer's data for propellers (1951 Transactions of the Institution of Mechanical Engineers, "Calculation and Measurement of Torsional-vibration Stresses in Marine Propeller Shafting"); but for the frictional damping at bearings, good reason is given for believing that damping forces absorb as much as 25 to 30 per cent of vibration energy per cycle, instead of the widely used figure of 5 per cent energy loss.

On the instrument side, the construction of a frequency modulated torsiograph is described, which, as it has no element corresponding to the mass and spring combination of a seismic type, has no lower limit. It has, however, a distinct upper frequency limit, which can be controlled by the method of construction.

Examples of the analysis of both propeller-excited vibrations and of gearing-excited vibrations are given.

In a paper presented to a joint meeting of the Institution of Naval Architects and Institute of Marine Engineers, in February last, and entitled "Practical Approach to some Vibration and Machinery Problems in Ships," by Mr. T. W. Bunyan, B.Sc. (Eng.), the author points out that vibration of hulls and machinery is by far the most common form of trouble submitted for investigation to the Engineering Research Department of Lloyd's Register of Shipping. Many examples are quoted, together with the appropriate action necessary to rectify the trouble, and design stage recommendations are made. Of particular interest is a method of tuning-out an engine-excited hull vibration by means of ballasting (only possible with tankers). In an actual case, the hull critical frequency was reduced from 102 to 86 cycles per minute.

While the propeller itself, and the lines of the after body and of the rudder, are very important factors influencing propeller forces, it is a matter of the author's experience that most ships which are subject to severe propeller-excited vibration have small aperture clearances. Suggested minima for such clearances are:—0.15 dia. for leading edge to stern-frame; 0.08 dia. for trailing edge to rudder-stock (or post); 0.08 dia. for tip to bottom of stern-frame; and 0.10 dia. for tips to arch of stern-frame. As power and propeller diameter increase for single-screw ships, these clearances will tend to be pushed to the limit. An appropriate formula is given for estimating the force required to produce an acceleration of 1 ft. per second per second at the stern of a vessel when on resonance.

In discussing the ill-effects of both over-and-under-tightening bolts used in dynamic assemblies, a neat and robust arrangement is described where, by means of a poker gauge and micrometer, a close assessment of the bolt load can be made while tightening up.

A thorough analysis is made of the causes and the cure of a rare type of top-end failure, the symptom of which is that, white-metal is progressively hammered out of the forward and after crosshead bearing—usually after remetalling.

Salford Electrical Instruments, Ltd., are manufacturing a new ⅓-octave spectrometer, originally developed for the Admiralty Engineering Laboratory. It can resolve the whole spectrum from 10 cycles per second to 100 kilocycles per second, and can be used for a variety of applications. The high-frequency range makes it particularly suitable for the analysis of noise emitted by jet engines, and it has also been used for a variety of work, from production testing of transformers and amplifiers to the analysis of vibra-

tions in turbine blading and rotating shafts. In addition, it has been successfully used in analyzing and measuring the torsional vibrations for marine Diesel machinery.

CORROSION PROBLEMS

Sea-water circuits in vessels making frequent calls in port can become rapidly fouled-up, due to marine growths. The most prone to such fouling is the sanitary system, due to the comparatively small diameter of piping used. In the *Enotria*, an anti-fouling unit has been installed, by means of which the sea-water circuits (fire-fighting and sanitary) can be chlorinated. The necessary chlorine is generated on board by the low-voltage electrolysis of a brine solution in a compact chlorinating unit, complete with tank, pumps and controls. An ejector serves to mix the chlorine, under pressure, into solution with a carrier stream of sea-water. The chlorinated water is then fed into the main sea-water circuit at a suitable rate, the only maintenance necessary being the daily test of the concentration of active chlorine.

An initial chlorination for a total of 27 hours in 45 days resulted in a general improvement throughout the system; considerable numbers of dead mussels were also discharged. When the treatment was increased to 288 hours in 75 days (with a maximum consecutive period of treatment of 45 hours), the system was completely cleared and maintained clear, even in the hot seasons. Chlorination should be carried out all the time a vessel is in port.

Tests show that the *p*H value of chlorinated sea-water is about 8, the slight alkalinity protecting such materials as copper or aluminum in the systems.

A claim is made for Alumicoat that it successfully protects mild steel and low-allow steels in the presence of corrosive media, and, in particular, that it also affords protection against high-temperature oxidation. Its effectiveness has been illustrated in the case of the economizer of a ship's boiler, which had required replacement parts at frequent intervals before treatment was applied, but which has required no replacements since treatment.

The part to be protected is first thoroughly cleaned, fluxed in a salt bath (patented mixture) and then dipped in a bath of the molten protective material, either pure aluminum or one of its alloys. The coating penetrates the steel to a depth of 0.0015 in. to 0.002 in., and approximately 0.0005 in. to 0.001 in. remains as a protective coating on top of the surface, the actual thickness being determined by the material used and the duration of the immersion.

It may make possible the use of mild or low-alloy steels in place of expensive alloys for certain parts of jet and gas-turbine machinery, and should certainly be useful for protecting parts of boilers and Diesel machinery.

The corrosion of metal parts and sub-assemblies carried as cargo can be successfully combated by anti-corrosive treatment and the effective use of inhibitors. However, in the case of motor cars, it is reported that there is serious damage, due to mold attack on coach-work and upholstery, together with blistering of external paintwork. The blistering requires the control of hold temperatures and humidity, and the increased use of mechanical ventilation and air-drying will greatly assist in this direction. The mold attack is now being prevented by the use of vapor-phase inhibitors.

A recently developed vapor-phase corrosion inhibitor is cyclohexylamine carbonate (or CHC), which is a solid vaporizing at atmospheric temperatures, and acting as a powerful corrosion inhibitor. To be fully effective, the vapor loss should be low, so that it is particularly suitable for impregnating paper which can be used in packaging the objects to be protected. Cast-iron, mild steel, chromium-plated steel, tinplate, aluminum, zinc and solder have all been protected, even when exposed to 104 deg. F. in a water-saturated atmosphere for two weeks, and it has proved highly effective, even in the presence of salt-water contaminated moist air under the same conditions. If introduced after attack has commenced, CHC arrests the process. Under certain conditions, it has been found that it has some corroding influence on copper, brass and magnesium.

Vapor-phase inhibitors are cleaner and easier to handle than greases and oils. They do not require subsequent removal, and they are particularly suitable for the protection of components of intricate construction, as the vapor diffuses even to the most inaccessible sections.

The results of the experimental controlled-current cathodic protection in the hull of H.M.C.S. *New Liskeard* against corrosion indicate that, while it is possible to extend the period between dry-docking to two years, due to improved corrosion protection, the fouling growth becomes too severe to allow a two-year period out of dock, unless great care is taken in choosing the quality of the paint and in the pre-treatment of the hull surface.

A patent has been registered, claiming to reduce the fouling of ships' hulls by marine growths and organisms by transmitting ultrasonic frequency vibrations through the hull. A transducer is arranged, so that each end of its coil is rigidly attached to the hull by means of curved projections. More than one can be attached, if necessary; each unit comprises a step-up transformer (supplied with 230 volts A.C.), a rectifier and an oscillator which feeds the transducer winding.

MATERIALS

During the past few years, a great deal of effort has been put into investigations concerning the cause of brittle fracture of steel, and efforts have been made to determine the criterion governing it, and to devise a simple laboratory test for assessing the properties of the steel in comparison with the criterion.

In the Fifth Progress Report of the Ship Structures Committee (British), an analysis is made of the relevant data concerning structural failures of almost 100 ships. Special attention is given to the correlation, or to lack of correlation, between the service experience and laboratory examinations and tests. No relation is

found between service results and laboratory tensile tests of the steels. However, Charpy tests show that those plates in which fracture originated are generally more notch-sensitive than those in which fracture did not originate. An analysis is made of the notched-bar test data, and of the relations between service experience, notch sensitivity and chemical composition of the steels concerned. In every fracture examined, the starting point could be traced to a point of stress concentration due either to design detail, accidental damage, welding defects, or the metallurgical effects of arc strikes, or of fabricating operations such as welding, flame cutting or cold forming.

The Fifth Progress Report of the United States Ship Structures Committee is primarily concerned with a study of the influence of deoxidation and chemical composition on the properties of the ship steel. In particular, an examination of results of standard keyhole Charpy tests and U.S. Navy tear tests shows that the transition temperature, as indicated by the one test, does not correlate very well with that indicated by the other test; so that, estimates of transition temperature become difficult.

"The Geometrical Size Effect in Notch Brittle Fractures" was discussed by Mr. A. A. Wells, B.Sc. (Eng.), Ph.D., in his paper of this title, presented to the North-East Coast Institution of Engineers and Shipbuilders, in March last. The behavior of a large prototype can never be accurately predicted from structural-strength investigations on a smaller-scale model, and this scale effect is dependent upon geometrical and metallurgical differences between prototype and specimen. In order to eliminate the metallurgical variable, all experimental test-pieces are taken from a single plate of steel, sufficiently thick for all sizes of specimen to be cut from immediately adjacent portions.

From a theoretical analysis of a cracked stressed plate, the author shows that the rate of release of strain energy increases as the crack extends. It is thus postulated that there is a critical crack length, which must be exceeded before the crack will run naturally at high velocity under the sole influence of elastic energy released from the plate. For a plate thickness of unity, the energy expended at the surface of the crack for each unit increase in length of crack is called its surface energy (S); thus, the rate of release of strain energy must be greater than S for the crack to be self-propagating.

The whole of the energy released during propagation of a crack is first converted into kinetic energy, and, finally, through rapid damping of vibrations, it is converted into heat, the change taking place in a few milli-seconds. The author describes a method of obtaining experimental values for surface energy using thermo-couples, and the results obtained compare satisfactorily with calculated values.

Tests similar to Docherty's test are described for slow-bend specimens (all geometrically similar, but varying in size), surface energies being both measured and calculated for specimens near the critical size, for transition from brittle to ductile fracture. The temperature is reported as being 68.5 deg. F. $\pm$0.2 deg. F. In the four smallest specimens, brittle fracture is suppressed altogether, but, with increasing size, the percentage of brittle area also increases. The brittle specimens show alternating brittle and ductile areas, as the fracture appears to be stopped by strain-energy exhaustion, and is then restarted.

The author concludes with particular reference to brittle fractures, which have occurred in full-sized structures, and gives reasons why residual stress relief can be so beneficial where the presence of notch-brittle material in a structure is suspected.

An investigation into the occurrence of extensive fractures in welded merchant ships was begun at Cambridge in 1943, on behalf of the Admiralty Ship Welding Committee, and has been continuing under the aegis of Professor J. F. Baker, O.B.E., Sc.D., and Mr. C. F. Tipper, Sc.D. A report presented in the form of a paper entitled "The Value of the Notch Tensile Test," was read before the Institution of Mechanical Engineers, in October last. The authors put forward the considerations which have led to their proposal of a notched tensile test. Such a test is claimed to give a reliable indication of the service performance of the material.

A quantitative criterion of brittleness is difficult to obtain. Thus, it is proposed that the presence of a cleavage-type fracture is the only sure indication of brittleness, and that energy to fracture, ductility or strength can give supporting evidence and a measure of the brittleness. By experiment, it has been found that a simple notched bar, pulled in tension at a suitable temperature, produces fractures resembling service fracture in every respect. It is important that the test-piece must be of full plate thickness (for geometrical and metallurgical reasons, as indicated in Mr. A. A. Wells' paper), the width-thickness ratio should not be less than 2 to 2½ whenever practicable. Two notches are machined in the narrow sides of the test-piece, half-way down, the notch depth being ⅛ in., while the total angle is 45 deg., and the bottom radius 0.25 mm.

In describing the method of carrying out the test, the samples are warmed or cooled to a temperature above or below that at which it is desired to break the specimen. After insertion in the grips of the machine, a thermo-couple is attached by means of a spring clip. The loading is begun according to the rate of rise or fall in temperature, and, at such a rate, that fracture occurs at the required temperature. No significant difference is found if the specimen is immersed in a bath at constant temperature, or if the rate of loading is varied. By such means, the temperature at which transition from crystalline to fibrous fracture occurs can be readily determined. The transition is always accompanied by a sudden change in ductility. The temperature of transition does not usually vary with the direction of rolling and this agrees well with service failures, which, in many instances, traverse plates in an irregular manner, irrespective of rolling direction.

Comparison between notch tensile-test results and

corresponding Charpy and Izod results, indicate the former to be a much surer guide to service behavior, as service fractures have actually propagated when energy values are widely different, while the notch tensile test correctly forecasts that fracture would be brittle in each instance.

Certain criticisms of the test are discussed, and it is finally concluded that the simple notch tensile-test bar, tested as described, can be a reliable test of service performance. Not only does it give direct correlation with service, but it also may be used to predict the behavior of a material in a more complicated form.

A series of experiments undertaken by Messrs. Burmeister and Wain, and by various Danish testing laboratories on their behalf, to determine the fatigue strength of Diesel-engine parts up to either full-scale, or at least to a comparatively large scale, is dealt with by Mr. P. E. Wiene in a paper entitled "Full-scale Fatigue Tests of Diesel-engine Elements," which was read before the Institute of Marine Engineers, in November last.

When considering fatigue testing, it is not always easy to see the correlation between, say, Wöhler tests on small specimens and experience in practice. However, small rotary bend tests can be used for making qualitative assessment of varying heat-treatments and surface conditions. The results of such tests indicate clearly the great benefit to be obtained by nitriding a suitable steel. As is well known, most steels (which have not been surface treated) show a marked notch sensitivity, the notched specimen having a bending fatigue strength of about 50 per cent of that of a plain specimen. The effect of nitriding is to raise the strength of the plain specimen, and this same strength is fully maintained when the specimen is notched; thus, on the size of specimen tested, the effect of nitriding is to raise the notched bending strength by some 300 per cent and, at the same time, to provide a wear resistant surface. Small crankshafts are sometimes nitrided over the complete surface. Such a favorable effect is probably due to the fact that nitrides occupy a larger volume than the original iron used in their formation, thus setting up a powerful compression stress at the surface.

Spring steel is another material showing considerable notch sensitivity, the effect of rough turning being to reduce the bending-fatigue strength by about 33 per cent. It is probable, therefore, that the most important criterion for spring material is not exact conformation to analysis, but simply that the surface should be faultless, that cracks are avoided and that decarbonization is absent. For spring steels, which, of course, are not suitable for nitriding, the same beneficial effect can be induced by the mechanical means of shot-peening the material. The bending-fatigue strength in the smooth condition is raised, and, in the rough-turned condition, it is raised by 100 per cent to a figure about halfway between that for smooth peened condition and the smooth non-peened condition. Shot-peening after heat-treatment has resulted in a noteworthy reduction in the number of spring fractures during the past four years.

Tests on a 0.2 per cent carbon steel indicate that there is a sharp decrease in bending-fatigue strength, as the fillet radius at the bottom of the notch is reduced from 3.0 mm. to 0.1 mm. Comparison of the results with a specimen three times the size and tested by a push-pull arrangement (B.S.R.A. experiment), but with practically the same root fillet, show that increase in scale, together with, perhaps, the change of test procedure, have reduced the fatigue strength by practically 45 per cent.

A series of tests are reported which were carried out on full-sized small steel crankshafts (56-mm. diameter pin) in a pulsator fatigue-testing machine, and the use of a re-entrant fillet radius of 8 mm., at both pin web and journal web junctions, is found to give a marked increase in strength. For cast-iron crankshafts, the improvement is even more marked, and the same safety margin can be achieved as with normal mild-steel shafts.

Of considerable interest are the tests on the piston rod-crosshead assemblies, carried out with scale models in which the maximum piston-rod diameter is only 36 mm., owing to the limitation of the pulsator machine used. The effect of repeatedly pre-tightening the nut is reported, as is also the effect of variation of thread design. Maximum improvement is obtained by reducing the radial height of the piston-rod threads (thus reducing the bending stress on them) and compensating the resulting increased stresses in the nut threads by using an improved steel for the nuts and by rolling the threads, followed by careful attention to pre-tightening procedure.

MISCELLANEOUS

Tailshaft Investigation

A further report has been issued by the American Society of Naval Architects and Marine Engineers, in the form of a paper entitled "Further Results of the Society's Investigation of Tailshaft Failures," by Mr. H. R. Neifert and Mr. James H. Robinson, which was read before the Society in November last. One phase covers reversed-bending fatigue testing of ¼-scale T-2 tanker tailshaft-propeller-hub assemblies; the other, full-scale tailshaft bending-stress measurements made at sea on a "Victory" ship.

The model tailshafts were 5¾ in. in diameter—a size sufficiently large to minimize the influence of size effect on the fatigue tests, the results of which can be summarized as follows:—

(1) The maximum, allowable nominal calculated stress for the original model tailshaft is 10,000 lb. per sq. in.

(2) The use of relief grooves at the forward end of the propeller boss and the after end of the sleeve does not increase the fatigue resistance against breaking-off across the taper section inside the boss.

(3) Surface rolling the taper increases the allowable stress to prevent fracture across the taper portion, to 17,000 lb. per sq. in., an improvement of about 70 per cent.

For the actual measurements of shaft strains in a ship at sea, the tailshaft was bored with a 4-in. diameter central hole, and leads from 16 strain gauges were brought out to slip rings inside the ship. The gauges were mounted in pairs at 45-deg. intervals round the shaft. Each set of four gauges, comprising a pair at each end of a diameter, was connected as the four active arms of a Wheatstone bridge. In order to calculate the stresses from a knowledge of the hydrodynamic forces acting on the propeller, wake surveys were carried out, in the David Taylor Model Basin, for the two conditions of draft it was proposed to run at, and, from the survey results, the forces acting were calculated and thus tailshaft stresses were deduced.

The comparison of measured stresses with computed values is not so close as was hoped for, the computed values being consistently low, only about 60 to 70 per cent of the measured values. On the other hand, it can be argued that, to have got even as close as this, with a computation based only on a tank test for wake distribution, is indeed remarkable. The work is to continue.

Spur-teeth Stresses

In a paper entitled "Bending Stresses in Spur-gear Teeth—Proposed New Design Factors Based on a Photo-elastic Investigation," by Mr. M. A. Jacobson, which was presented to the Institution of Mechanical Engineers, in April last, the author is concerned with the photo-elastic measurement of stresses in spur-gear teeth and their interpretation, in order to estimate revised strength factors for the British Standard Specification on the design of spur gears. He also attempts a correlation between his own results and the fatigue test results of other investigators. The strength is investigated of typical teeth generated by standard cutters, conforming to the British Standard basic rack (20-deg. pressure angle), with the object of deriving maximum safe bending loads for involute spur gears. Strength factors for such spur gears are given in the form of charts, covering the whole range of spur-gear combination, both with and without addendum corrections. A list of service factors is given, as well as a graph correlating bending-fatigue data for three classes of surface finish, *viz.*, ground, polished and hobbed.

Tanker Cleaning

The Tyne Tanker Cleaning Co., Ltd., commenced operations early in the past year. They are strategically placed and have the facilities for the reception and treatment of contaminated ballast, tank washings and oily sludge, and for gas-freeing of tankers visiting the port. The plant consists of two reinforced-concrete separators, fitted with electrically-operated automatic skinning apparatus, each with an output of 500 tons per hour, and a reception tank of 2,000 tons capacity for tank washings and oily sludge. Further tanks are arranged for the treatment and dehydration of the oily sludge. Two Scotch boilers provide ample steam for the plant requirements, and steam can also be piped on board, if necessary.

This is a joint venture of the major oil-tank ship-repairing firms on the River Tyne and of the Tyne Improvement Commission, the major British oil companies and owners of oil-tank ships having given valuable technical assistance in the design of the installation. Such a combined effort is to be wholeheartedly applauded, as it helps keep the seas and our coastline clean and free from tank washings, it shortens the tankers' stay in port for repairs and, in so doing, helps owners and attracts more trade to Tyne ship-repairs.

GEORGE LEPPERT

NUCLEAR REACTOR DESIGN THEORY

THE AUTHOR

was born in 1924 and graduated from the University of Wisconsin in 1947. He served in the United States Navy from 1943 to 1946 and is a qualified submarine officer. He received the M.S. (1952) and Ph.D. (1954) degrees in Mechanical Engineering from Illinois Institute of Technology and was engaged in the Reactor Engineering Division of Argonne National Laboratory from 1950 to 1953. Since 1954, he has been in the Department of Mechanical Engineering of Stanford University, teaching courses in heat power and nuclear engineering and conducting research in reactor heat transfer.

THE DISCOVERIES of modern physics have provided the possibility of obtaining large quantities of energy from the fission of heavy isotopes such as uranium-235, uranium-233 and plutonium-239. As has been discussed in some detail in recent articles in this journal (3, 5), the capture of a neutron by a nucleus of one of these fissionable materials will usually result in the production of two light nuclei, two or three more neutrons, and a relatively large amount of energy which is readily converted to heat. Developing economical means for the production of useful power in a controlled manner constitutes an important part of the new branch of technology which is called nuclear engineering.

The engineering design of a nuclear reactor is based on an understanding of the nature of the fission process and, to a great extent, on predicting the behavior of the neutrons. It will be recalled that the requirement for *criticality* is just that the neutron population be constant as the chain reaction proceeds. In order to design for such a condition, the engineer must know the average number of neutrons produced per fission, their kinetic energy at "birth," and their probable loss or utilization in the various processes which they may undergo before they are ultimately lost to the reaction. These processes include leakage from the reactor core; resonance capture, which is capture by fissionable nuclei which does not lead to fission; parasitic capture, which is capture by materials in the core other than fuel—i.e., moderator, structural elements, coolant, control rods and other poisons; and, of course, fission capture, which is the process which permits the reaction to continue.

Nuclear reactors may be categorized as *thermal, intermediate,* or *fast* according to the average energy of the neutrons which produce fission. The thermal reactor design, in which the average neutron kinetic energy is reduced to a very low value (about 1/40 electron-volt) in the moderator before the neutrons cause fission, is the most common. Practically all of the theory which will be reviewed here will pertain to this type.

Fast reactors are designed with no attempt to slow the neutrons, although some slight moderating effect is produced by the coolant and other nuclei in the core. The average neutron energy in a fast reactor is around 0.2 mev, at which energy all cross-sections are very low. The small values of capture cross section at high neutron energy permit the use of structural materials and coolants which would "poison" a thermal reactor. On the other hand, the fission cross sections are also lower and a very high neutron flux is required for the production of useful power.

A very important advantage of the fast reactor is its adaptability to breeding, which is brought about by the decrease in capture cross section relative to fission cross section. Another important advantage of fast and intermediate reactors is that they are poisoned to a much lesser degree by the accumulation of fission products. This is an important problem with thermal reactors, since certain of the fission products (notably xenon and samarium) have very large capture cross sections for thermal neutrons.

Many other engineering problems are of major importance to reactor design, including shielding of the surroundings against radiation damage, selection of corrosion-resistant materials and design for optimum heat transmission. Discussions of these will be deferred for succeeding articles.

MULTIPLICATION FACTOR

Definition of k

The condition of criticality has been shown to be expressible in terms of a *multiplication factor*, k, which is defined as the ratio of the number of neutrons of any one generation to the number of the preceding generation. For an infinite medium, where there is no leakage of neutrons from the system, the multiplication factor may be expressed as follows

$$k = \eta \varepsilon p f \qquad [1]$$

where η is the average number of fast neutrons emitted per thermal neutron captured

ε is the fast fission factor
p is the resonance escape probability
f is the thermal utilization factor.

Furthermore, if the system is finite, the neutron balance must consider the probability of some neutron leakage to the surroundings. The *nonleakage probability*, P, is defined such that the condition for criticality is

$$k P = 1 \qquad [2]$$

and the product, k P, is called the *effective multiplication factor*, k_{eff}.

Variation of η and ϵ

Values of η for both thermal and fast neutrons are given in the following table for various fuel materials (1):

Material	Thermal Neutrons	Fast Neutrons
U-235	2.11	2
Pu-239	1.95	2
Natural Uranium	1.32	1

These values are characteristics of the fuel and do not depend upon the arrangement of the fuel in the core.

The other three factors in the four-factor formula do depend, to some extent at least, on the arrangement of fuel and moderator. A distinction may be made in this regard between a *homogeneous reactor*, in which the fuel and moderator are intimately mixed, and a *heterogeneous reactor*, in which the fuel is lumped in definite aggregates, such as rods or plates, and is separated by collections of pure moderator.

The fast fission factor ϵ depends on the probability that a fast neutron will produce fission before it is slowed down. The chance for this is practically nil in a homogeneous reactor, since the fast neutrons are in moderator material as soon as they are produced. Consequently, ϵ is unity for the homogeneous arrangement, and there is no increase in k from fast fissions.

In a heterogeneous reactor, however, there is a finite probability that a fast neutron may cause fission before it escapes from the fuel rod in which it is produced. The larger the rod size, the greater this probability becomes, but the variation of ϵ, and therefore its effect on the multiplication constant, is not large. Figure 1 shows this variation in natural uranium fuel rods of various diameters. A value of 1.03 for the fast fission factor in natural uranium is a useful value to retain as a first approximation.

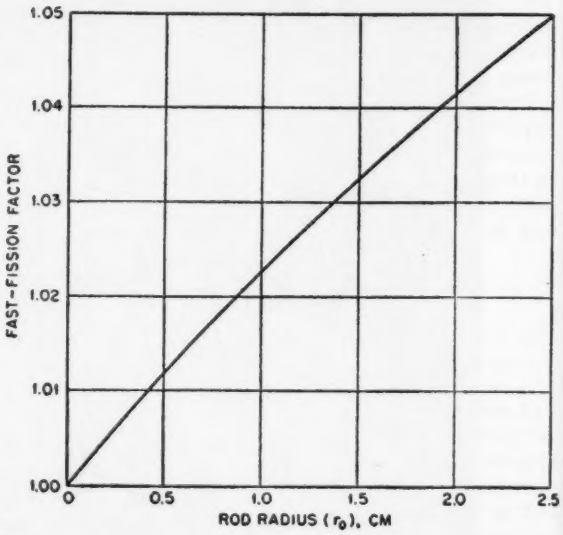


Figure 1. Fast fission factor in natural uranium fuel rods. (Reprinted with permission from Principles of Nuclear Reactor Engineering by S. Glasstone, 1955, Van Nostrand Company.)

Variation of p and f

Once the choice of fuel composition has been made, the value of k is determined largely by the product of resonance escape probability, p, and thermal utilization factor, f, since η is determined entirely by the nuclear properties of the fuel and ϵ varies only slightly with fuel rod diameter. Consequently, one design problem for such a reactor may be stated as that of obtaining the maximum possible value of the product, pf.

Considering these factors in a qualitative way, it may be seen that the resonance escape probability will increase as the ratio of moderator to fuel increases, since the greater proportion of moderator nuclei will increase the likelihood of the neutron being slowed below the resonant energies before resonant absorption can take place.

On the other hand, the thermal utilization factor will *decrease* as the ratio of moderator to fuel increases, because f by definition is the fraction of thermal neutrons absorbed by the fission process. As fuel becomes relatively scarce, thermal neutrons lost to non-fission processes constitute a greater proportion of the total. Values of these quantities are given below for uniform mixtures of natural uranium and graphite in various proportions (6).

Nm/Nu	p	f	pf	k
200	0.64	0.89	0.57	0.75
300	0.70	0.84	0.59	0.78
400	0.74	0.80	0.60	0.79
500	0.77	0.76	0.59	0.78

These values of k, obtained from the four-factor formula with $\epsilon = 1$ and $\eta = 1.32$, show that a chain reaction will not occur with a homogeneous mixture of graphite and natural uranium.

A further consideration in maximizing pf (hence k) arises in the heterogeneous reactor design from the effect of lumping the fuel, which is advantageous from the standpoint of its effect on the resonance escape probability. A certain proportion of the neutrons in the moderator will have energies near the resonant values for the fuel, and will therefore be lost by resonant absorption as the neutrons diffuse into a fuel rod. As the neutrons penetrate further into the fuel, however, the proportion of those at resonant energies decreases, and average loss to resonant capture decreases. Furthermore, the fuel is a very poor moderator, so that neutrons of greater than resonant energy which enter the fuel tend to return to the moderator at about the same energy. The probability is high that these neutrons will then be slowed below resonance before again entering a fuel lump.

Just as in the case of the variation of moderator to fuel ratio, however, the gain in p which results from lumping the fuel tends to be offset by a decrease in the thermal utilization factor. The thermal neutrons which diffuse into the fuel are absorbed in both fission and non-fission capture, with the result that the average neutron flux decreases toward the center of the rods, in a manner illustrated by Figure 2. Since the average thermal flux is then greater in the moderator than in the fuel, losses of thermal neutrons to the moderator itself and to various poisons become relatively more important, and the value of f decreases with increasing fuel lump thickness.

Figure 3 shows that the net effect of heterogeneity may be beneficial, however, since for a molar ratio of graphite to natural uranium of around 150, k is greater than 1.1 for all four fuel rod radii in the range of 0.5 to 3 cm, and a chain reaction is possible. The X-pile at Oak Ridge National Laboratory is a reactor of this type with fuel rod radius of almost 3 cm, and there are a number of other natural uranium-graphite reactors in operation. Most notable of such reactors was CP-1, the uranium-graphite "pile" built under the West Stands of the University of Chicago's football stadium. This reactor "went critical" on December 2, 1942, and demonstrated for the first time a self-sustaining nuclear chain reaction. The values of k in Figure 3 are to be compared with those given earlier for a homogeneous mixture of graphite and natural uranium, where the maximum value of k was seen to be about 0.8.

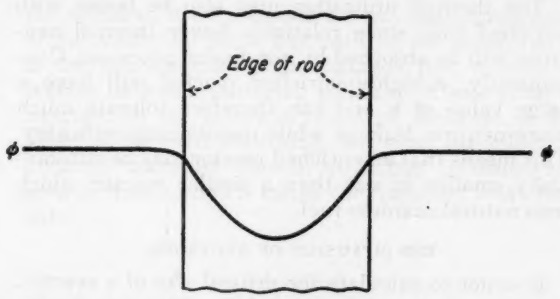


Figure 2. Typical thermal flux distribution in a fuel rod. (Reprinted with permission from Introduction to Nuclear Reactor Engineering by R. Stephenson, 1954, McGraw-Hill Book Company)

Fuel Enrichment

If fuel is available with more fissionable material than the 0.7 per cent U-235 which occurs in natural uranium, greater values of η and p may be obtained. The value of η depends on the average number of neutrons per fission, ν, and on the fuel cross sections, i.e.,

$$\eta = \frac{\Sigma_f}{\Sigma_a}\nu$$

so that a greater proportion of fissionable nuclei in the fuel results in a larger value of macroscopic fission cross section, Σ_f, for the fuel mixture. Consequently, η for pure U-235 is about 2.11 compared to 1.32 for natural uranium.

The resonance escape probability is also greater in an enriched reactor, because a smaller ratio of uranium to moderator will be needed for criticality. With the increased proportion of moderator nuclei, the probability increases that a neutron will be slowed to thermal energy before encountering a uranium nucleus, until p becomes almost unity for pure fissionable material. Since, for a heterogeneous reactor, ϵ is a little more than unity, it is approximately true that

$$k = 2.1\,f \qquad \text{for pure U-235.} \qquad [3]$$

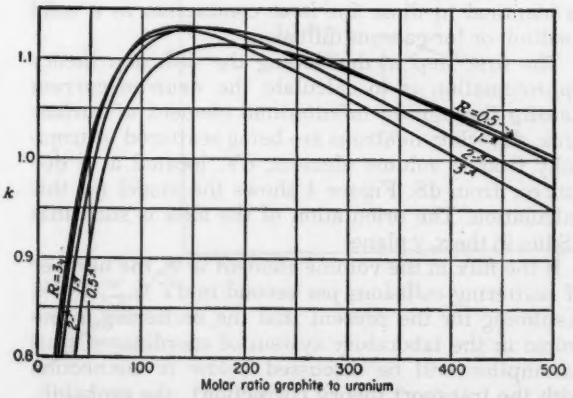


Figure 3. Multiplication factor for natural uranium-graphite reactor. Fuel rod radius R is given in cm. (Reprinted with permission from Introduction to Nuclear Reactor Engineering by R. Stephenson, 1954, McGraw-Hill Book Company)

The thermal utilization may also be better with enriched fuel, since relatively fewer thermal neutrons will be absorbed by non-fission processes. Consequently, a highly enriched reactor will have a large value of k and can therefore tolerate much more neutron leakage while maintaining criticality. This means that an enriched reactor may be substantially smaller in size than a similar reactor which uses natural uranium fuel.

THE DIFFUSION OF NEUTRONS

In order to calculate the critical size of a reactor, it is necessary to derive suitable mathematical relationships describing the neutron balance which was discussed above in a qualitative way. The basic equation, stated in words, is that the time rate of increase of the neutron population is equal to the rate of neutron production minus the rate of absorption and minus the rate of leakage, i.e.,

$$\frac{\partial n}{\partial t} = \text{Production} - \text{absorption} - \text{leakage} \qquad [4]$$

The absorption term may be written down readily as $\Phi\Sigma_a$, but in order to evaluate the leakage rate we shall need to use diffusion theory, and in order to evaluate the neutron production term we must consider the slowing down mechanism and Fermi age theory.

Diffusion Theory

In the theory which is about to be described, let us suppose that all neutrons have the same velocity and that the scattering collisions which take place alter the directions of the individual neutron velocities but not their magnitudes. These approximations are valid for thermal neutrons in a weakly absorbing medium such as a moderator. If the further assumption is made that the direction of a neutron velocity vector does not depend upon the location of the neutron in the medium, which is a good approximation at distances of 2 λ_s or more from sources, strong absorbers, or boundaries, then the *diffusion theory approximation* is applicable. It will be seen that the mathematical form of the diffusion law for neutrons is identical to those for heat conduction in a solid medium or for gaseous diffusion.

The first step in developing the diffusion theory approximation is to calculate the *neutron current density* through an infinitesimal element of surface area, dS, when neutrons are being scattered isotropically from a volume element, dV, located at a distance r from dS. Figure 4 shows the model for this calculation. The orientation of the axes is such that dS lies in the x, y plane.

If the flux in the volume element is Φ, the number of scattering collisions per second in dV is $\Sigma_s \Phi$ dV. Assuming for the present that the scattering is isotropic in the laboratory system of coordinates (this assumption will be discussed below in connection with the transport theory correction), the probability that a neutron will be scattered in a suitable direction to permit it to strike dS is just (dS) $(\cos\theta)/4\pi r^2$, which is the solid angle subtended by dS at dV.

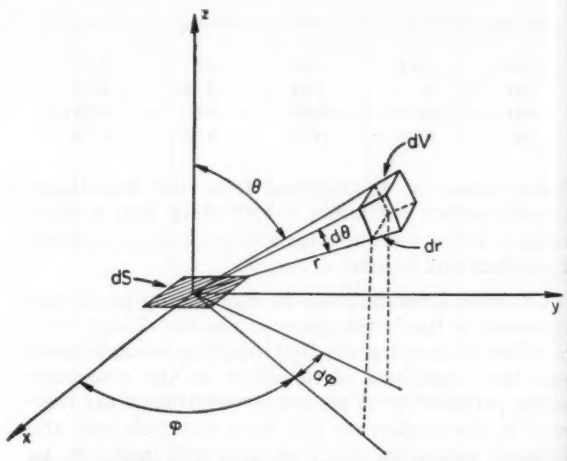


Figure 4. Physical model for the calculation of neutron current density. (Reprinted with permission from The Elements of Nuclear Reactor Theory by S. Glasstone and M. C. Edlund, 1952, Van Nostrand Company)

The number of neutrons which reach dS after scattering in dV is then

$$(\Sigma_s \Phi\, dV)\left(\frac{dS\cos\theta}{4\pi r^2}\right)(e^{-\Sigma_s r}) \quad ,$$

the last factor being necessary to take into consideration the probability that a neutron scattered in dV at the proper angle to enter dS may be scattered again while traversing the distance r. The probability of absorption is ignored because the theory postulates a weakly absorbing medium.

We now wish to find the neutron current density, J_-, which is defined as the number of neutrons crossing unit surface per second in a given direction, in this case the negative z direction. This may be found by integration of the above expression over the volume of positive z, i.e., over all positive r; over a variation in ϕ from 0 to 2π; and over a variation in θ from 0 to $\pi/2$. This integral becomes

$$J_- dS = \frac{dS}{4\pi}\Sigma_s \int_0^{\infty}\int_0^{2\pi}\int_0^{\frac{\pi}{2}} \Phi\, e^{-\Sigma_s r}\cos\theta\sin\theta\, d\theta\, d\phi\, dr$$

where dV has been replaced by $r^2 \sin\theta\, dr\, d\phi\, d\theta$.

In order to perform this integration, expand Φ in a Taylor series and retain only the first order terms

$$\Phi(x,y,z) = \Phi_0 + x\left(\frac{\partial\Phi}{\partial x}\right)_0 + y\left(\frac{\partial\Phi}{\partial y}\right)_0 + z\left(\frac{\partial\Phi}{\partial z}\right)_0$$

where the subscript "0" implies evaluation at the origin.

When x, y and z are expressed in spherical coordinates and Φ is substituted in the above integral, considerable simplification is possible. It can be shown that the terms in x and y make no contribution and that the neutron current density becomes

$$J_- = \frac{\Phi_0}{4} + \frac{1}{6\Sigma_s}\left(\frac{\partial\Phi}{\partial z}\right)_0$$

Similarly, integration over the volume of negative z yields the current density in the positive z-direction:

$$J_+ = \frac{\Phi_0}{4} - \frac{1}{6\Sigma_s}\left(\frac{\partial \Phi}{\partial z}\right)_0$$

and the net current density in the positive z-direction is the sum of J_+ and J_-, with due regard being given to the negative direction of J_-

$$J_z = J_+ - J_- = -\frac{1}{3\Sigma_s}\left(\frac{\partial \Phi}{\partial z}\right)_0 = -\frac{\lambda_s}{3}\left(\frac{\partial \Phi}{\partial z}\right)_0$$

It can be shown that the above approximation for Φ is justifiable when the change in $\partial\Phi/\partial z$ is negligible over a distance of two or three mean free paths (λ_s). This will be true except closer than 2 or $3\lambda_s$ to a concentrated neutron source, a strong absorber, or a boundary. Similar results may be obtained for J_x and J_y.

Transport Theory

The *compound nucleus theory* postulates two stages to a nuclear reaction. In elastic scattering, for example, the first stage occurs when an incident neutron strikes a stationary moderator nucleus and is absorbed. The second stage occurs when the compound nucleus disintegrates to form products identical to the reactants except for a redistribution of the kinetic energies. If this theory is valid, there is no preferred direction of neutron emission with respect to the center of mass of the compound nucleus.

The laws of mechanics tell us that the motion of the center of mass of the compound nucleus is the same as the motion of the center of mass of the original neutron and target nucleus considered as a system, and the same as the center of mass of the products considered as a system. Since the motion of the center of mass is uniform, we may establish it as a convenient frame of reference in which scattering is isotropic. Experiments indicate the validity of this postulate of spherically symmetrical scattering in the center of mass system for neutrons of energies up to a few mev.

Although scattering is isotropic in the *center of mass* frame of reference, we wish to make our analysis on the basis of motion relative to our reactor, which is in the *laboratory* frame of reference. If we define the *scattering angle* as the angle between the directions of neutron motion before and after a scattering-type reaction, it is not difficult to show that

$$\cos \Psi = \frac{A \cos \theta + 1}{\sqrt{A^2 + 2A \cos \theta + 1}}$$

where

Ψ = scattering angle in laboratory system
θ = scattering angle in center of mass system
A = atomic weight of nucleus

For large values of A, the scattering angles approach equality, but for small A the scattering is strongly in the forward direction. By *forward scattering* we mean that scattering is non-isotropic in the laboratory system, with the preferred direction of the neutron velocity after scattering being in the general direction the neutron was travelling before the collision. If we consider the physical significance of the *scattering mean free path*, λ_s, we recall that it is the average crow-flight distance that a neutron travels between scattering collisions if scattering is isotropic in the laboratory system. Because of the effect of forward scattering, however, the effective distance travelled betwen collisions is somewhat greater than λ_s, and is called the *transport mean free path*,

$$\lambda_{tr} = \lambda_s + \lambda_{tr}\,\overline{\cos \Psi}$$

Solving for λ_{tr} yields

$$\lambda_{tr} = \frac{\lambda_s}{1 - \overline{\cos \Psi}} = \frac{1}{\Sigma_{tr}}$$

where $\overline{\cos \psi}$ denotes the average value of $\cos \psi$. The diffusion theory approximation developed earlier may be improved by replacing Σ_s by Σ_{tr} to account for non-isotropic scattering, i.e.,

$$J_x = -\frac{\lambda_{tr}}{3}\left(\frac{\partial \Phi}{\partial x}\right)_0$$

$$J_y = -\frac{\lambda_{tr}}{3}\left(\frac{\partial \Phi}{\partial y}\right)_0$$

$$J_z = -\frac{\lambda_{tr}}{3}\left(\frac{\partial \Phi}{\partial z}\right)_0$$

Neutron Leakage

The expressions which have been derived for the neutron current density may now be used to evaluate the leakage rate term previously mentioned. Figure 5 shows an infinitesimal volume element dV with

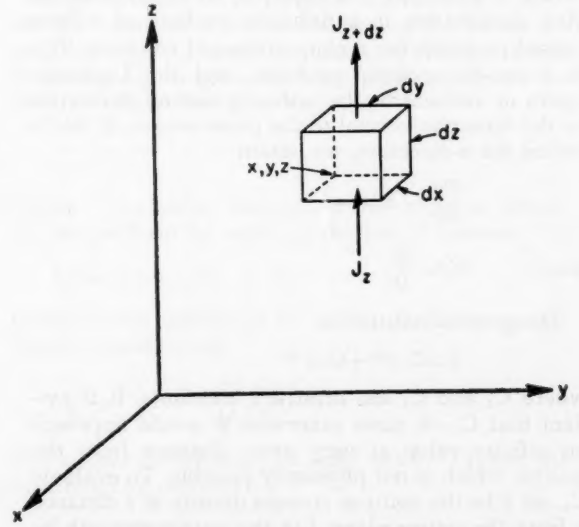


Figure 5. Physical model for the calculation of neutron leakage. (Reprinted with permission from The Elements of Nuclear Reactor Theory by S. Glasstone and M. C. Edlund, 1952, Van Nostrand Company)

net current densities J_z entering the lower face and J_{z+dz} leaving the upper face. The net flow of neutrons per unit time in the positive z-direction is

$$(J_{z+dz}-J_z)\,dx\,dy=-\frac{\lambda_{tr}}{3}\left[\left(\frac{\partial\Phi}{\partial z}\right)_{z+dz}-\left(\frac{\partial\Phi}{\partial z}\right)_z\right]dx\,dy$$

But $$\left(\frac{\partial\Phi}{\partial z}\right)_{z+dz}=\left(\frac{\partial\Phi}{\partial z}\right)_z+\left(\frac{\partial^2\Phi}{\partial z^2}\right)_z dz$$

Hence the net leakage rate in the z-direction is

$$-\frac{\lambda_{tr}}{3}\frac{\partial^2\Phi}{\partial z^2}$$

and in all three directions,

$$\text{leakage}=-D\left(\frac{\partial^2\Phi}{\partial x^2}+\frac{\partial^2\Phi}{\partial y^2}+\frac{\partial^2\Phi}{\partial z^2}\right)=-D\nabla^2\Phi$$

where ∇^2 is the Laplacian operator (defined for cartesian coordinates by the above equation) and D is the *diffusion coefficient.*

The Diffusion Equation

The diffusion equation [4] may now be expressed as follows

$$\frac{\partial n}{\partial t}=S-\Phi\Sigma_a+D\nabla^2\Phi \qquad [4a]$$

For steady state conditions, $\partial n/\partial t=0$, and in a medium without fissionable material, S, the production term, is also zero, and the equation simplifies to

$$\nabla^2\Phi-\frac{\Sigma_a}{D}\Phi=0$$

The boundary conditions and the most convenient coordinate system to use with this differential equation depend on the particular physical situation to which it is applied. For example, let us calculate the flux distribution in a diffusion medium of infinite extent in which lies a plane source of neutrons. This is a one-dimensional problem, and the Laplacian operator reduces to the ordinary second derivative in the direction normal to the plane source. If this is called the x-direction, we obtain

$$\frac{d^2\Phi}{dx^2}-K^2\Phi=0$$

where $$K^2=\frac{\Sigma_a}{D}.$$

The general solution is

$$\Phi=C_1e^{Kx}+C_2e^{-Kx}$$

where C_1 and C_2 are arbitrary constants. It is evident that $C_1=0$, since otherwise Φ would approach an infinite value at very great distance from the source, which is not physically possible. To evaluate C_2, let J be the neutron current density at a distance x from the source plane. Let the source strength be Q neutrons/(sec)(sq cm) and observe that, because of symmetry about the source plane, half the neutrons emitted will travel in the positive x-direction. Then, at the plane x,

$$J=-D\frac{d\Phi}{dx}=DC_2Ke^{-Kx}$$

and as $x\to0$,

$$J=\frac{Q}{2}=DC_2K \quad \text{or} \quad C_2=\frac{Q}{2KD}$$

$$\Phi=\frac{Q}{2KD}e^{-Kx} \quad \text{for a plane source.}$$

A similar solution may be obtained from the diffusion equation for a point source of neutrons in an infinite, homogeneous diffusion medium. In this case, spherical coordinates are used in the Laplacian, and the differential equation becomes

$$\frac{d^2\Phi}{dr^2}+\frac{2}{r}\frac{d\Phi}{dr}-K^2\Phi=0$$

where r is distance from the source. The solution is

$$\Phi=\frac{Q}{4\pi rD}e^{-Kr} \quad \text{for a point source.} \qquad [5]$$

Diffusion Length

These solutions for the flux show that Φ is reduced by a factor e when it travels the distance 1/K, or

$$L^2=\frac{1}{K^2}=\frac{D}{\Sigma_a}=\frac{1}{3\Sigma_{tr}\Sigma_a}=\frac{\lambda_{tr}\lambda_a}{3}$$

The quantity L is called the *diffusion length* and is directly related to the mean distance travelled by a neutron from its source to its point of absorption by a nucleus, as will now be demonstrated.

The mean square "crow-flight" distance, $\overline{r^2}$, which a neutron travels from the source to the point of absorption may be defined by the following expression

$$\overline{r^2}\int_0^\infty(4\pi r^2\Sigma_a\Phi)\,dr=\int_0^\infty r^2(4\pi r^2\Sigma_a\Phi)\,dr$$

since $(4\pi r^2\,dr\,\Sigma_a\Phi)$ is the rate of absorption of neutrons in a spherical shell of thickness dr. The flux may be evaluated from the solution of the diffusion equation for a point source, and $\overline{r^2}$ is found from the defining relation above to be just $6/K^2$ or $6L^2$. Diffusion lengths for several moderators are given below (1):

Moderator	L, cm
Water	2.88
heavy water	100.
Beryllium	23.6
Graphite	50.0

SLOWING DOWN OF NEUTRONS

Diffusion theory is adequate to describe the behavior of thermal neutrons in moderator and to give a first approximation to the neutron leakage rate. We will now consider in greater detail the scattering process in which the fast neutrons, produced in fission, are reduced to thermal energy.

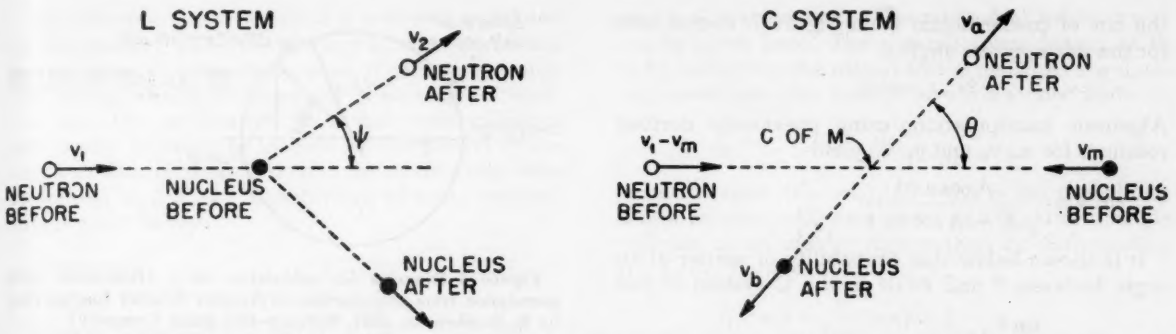


Figure 6. Neutron scattering in laboratory (L) and center of mass (C) systems. (Reprinted with permission from Principles of Nuclear Engineering by S. Glasstone, 1955, Van Nostrand Company)

Energy Changes

Mention has already been made of the spherically symmetrical (isotropic) scattering of neutrons referred to the *center of mass* (C) frame of reference, and to the convenience of the *laboratory* (L) frame of reference. Figure 6 shows diagramatically the collision of a neutron with a nucleus in both systems. In the L system, the neutron is initially moving to the right at velocity v, while the nucleus is assumed stationary. The latter assumption is justified because moderator nuclei, with only thermal energy, have negligible initial velocity. After the collision, the neutron is scattered at an angle ψ to its initial direction and with velocity v_2, while the nucleus rebounds in a different direction. The scattering angle, ψ, is of importance to our calculations, as is the kinetic energy loss ratio of the neutron, which may be expressed as E_2/E_1 or v_2^2/v_1^2 since $E=mv^2/2$.

In order to evaluate these quantities, let us study the C system in Figure 6, where the center of mass of the system consisting of neutron and nucleus is stationary. From the principle of conservation of momentum applied to the L system, we may equate total momentum of the system before and after scattering:

$$1 \cdot v_1 = (A+1)v_m$$
$$\text{or} \quad v_m = v_1/(A+1)$$

where A is the mass number of the nucleus, v_m is the velocity of the center of mass in the L system (unchanged by the collision), and the mass number of the neutron is unity. Now the initial velocities of the neutron and the nucleus in the C system are (v_1-v_m) to the right and v_m to the left, respectively, and the final velocities may be called v_a and v_b in that system. The scattering angle in the C system is called θ.

It is evident that the total momentum in the C system must always be zero. Before collision, the momentum of the neutron is

$$1 \cdot (v_1 - v_m) = v_1 - \frac{v_1}{A+1} = \frac{A\,v_1}{A+1}$$

to the right. The momentum of the nucleus is

$$A\,v_m = \frac{A\,v_1}{A+1}$$

to the left, and the total momentum is zero as predicted. After collision, the statement that total momentum is zero means that

$$1 \cdot v_a = A\,v_b$$

Writing the equation for the conservation of kinetic energy, we obtain

$$\tfrac{1}{2}(1)(v_1-v_m)^2 + \tfrac{1}{2}A\,v_m^2 = \tfrac{1}{2}(1)v_a^2 + \tfrac{1}{2}A\,v_b^2$$

The above expressions for (v_1-v_m), v_m and v_a may be substituted into this energy equation with the result

$$v_b = \frac{v_1}{A+1} \qquad \text{and } v_a = \frac{A\,v_1}{A+1}$$

These show that, in the C system, the particle velocities are unchanged by an elastic collision.

These results can be transformed into the L system by means of a vector addition of the center of mass velocity, v_m, to the final neutron velocity in the C system, v_a, to give the final neutron velocity in the L system, v_2. In vector notation,

$$\vec{v}_2 = \vec{v}_a + \vec{v}_m$$

Figure 7 is a vector diagram of this addition, which we can perform by applying the law of cosines

$$v_2^2 = v_m^2 + v_a^2 + 2v_m\,v_a\cos\theta$$

Using known values for v_m and v_a gives the *energy change in scattering*

$$\frac{E_2}{E_1} = \frac{v_2^2}{v_1^2} = \frac{A^2+2A\cos\theta+1}{(A+1)^2}$$

For a head-on collision, $\theta=\pi$ and $\cos\theta=-1$, and the maximum loss ratio is

$$\alpha = \left(\frac{A-1}{A+1}\right)^2 \qquad [6]$$

It is also possible at this point to relate the scattering angles in the two frames of reference. Writing

the law of cosines again from Figure 7, except now for the side opposite angle ψ,

$$v_a^2 = v_m^2 + v_2^2 - 2v_m v_2 \cos \Psi$$

Algebraic manipulation, using previously derived relations for v_a, v_m and v_2/v_1, yields

$$\cos \Psi = \frac{A \cos \theta + 1}{\sqrt{A^2 + 2A \cos \theta + 1}}$$

It is shown below that probability of scatter at an angle between θ and $\theta + d\theta$ in the C system is just

$$\frac{\sin \theta}{2} d\theta$$

Therefore the average value of cos Ψ may be found by integrating the product of cos Ψ and this probability, i.e.,

$$\overline{\cos \Psi} = \int_0^\pi \cos \Psi \frac{\sin \theta}{2} d\theta$$

$$= \int_0^\pi \frac{(A \cos \theta + 1)}{\sqrt{A^2 + 2A \cos \theta + 1}} \frac{\sin \theta}{2} d\theta$$

This integral may be evaluated by an elementary substitution to give

$$\overline{\cos \Psi} = \frac{2}{3A}$$

Average Energy Decrement

It has just been shown that the fractional energy change in a scattering collision depends only on the atomic weight of the scattering nucleus and on the scattering angle, θ, in the C system. If isotropic scattering is assumed about a point P which is at rest in this system, the probability of scattering into any solid angle is just the ratio of the solid angle to the total solid angle around the point, which is 4π. Figure 8 shows an imaginary sphere of unit radius around the point P, and we wish to find the number of neutrons which will be scattered at an angle between θ and $\theta + d\theta$ to the original neutron direction. The spherical surface intercepted between unit

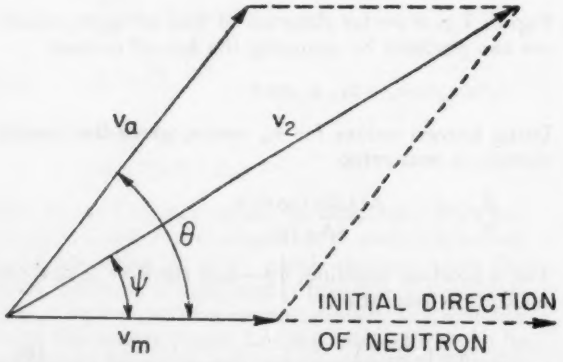


Figure 7. Vector diagram for transformation between C and L systems. (Reprinted with permission from Principles of Nuclear Reactor Engineering by S. Glasstone, 1955, Van Nostrand Company)

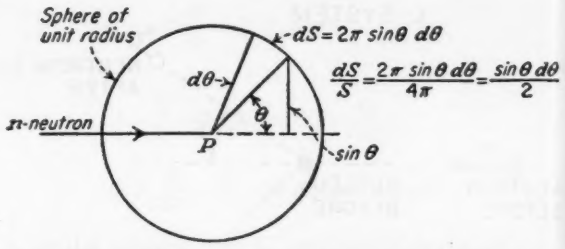


Figure 8. Diagram for calculation of ξ. (Reprinted with permission from Introduction to Nuclear Reactor Engineering by R. Stephenson, 1954, McGraw-Hill Book Company)

radii which rotate around the forward direction at angles θ and $\theta + d\theta$ is

$$dS = 2\pi (1) \sin \theta \, d\theta$$

and the probability of scatter into dS is

$$\frac{dS}{S} = \frac{2\pi \sin \theta \, d\theta}{4\pi} = \frac{\sin \theta \, d\theta}{2}$$

In studying the slowing down of neutrons, it is useful to define the *average energy decrement* as the natural logarithm of the energy change in one collision, i.e.,

$$\xi = \overline{\ln (E_1/E_2)} = -\overline{\ln (E_2/E_1)}$$

The average energy decrement is evaluated by integrating over all possible values of θ the product of $-\ln (E_2/E_1)$, which is known as a function of A and θ, times the probability of scatter into any angle θ, which was just derived with the aid of Figure 8.

$$\xi = \int_0^\pi -\ln \left[\frac{A^2 + 2A \cos \theta + 1}{(A+1)^2} \right] \frac{\sin \theta \, d\theta}{2}$$

This integration can be performed by straightforward methods to give

$$\xi = 1 + \frac{\alpha}{1-\alpha} \ln \alpha$$

where $\alpha = \left(\frac{A-1}{A+1} \right)^2$

For A>10, a good approximation is

$$\xi = \frac{2}{A + \frac{2}{3}}$$

We can now calculate readily the number of collisions to thermalize a neutron in a given moderator. For a 2 mev neutron in graphite, for example, if we take thermal energy as 1/40 ev, the total energy ratio is $2 \times 10^6 / 0.025$ or 8×10^7. The total logarithm reduction is just the natural logarithm of this number, or 18.1, while the logarithmic reduction in energy per collision, ξ, is about $2/(12 + \frac{2}{3})$ or 0.158 for carbon. Hence 18.1/0.158 or 114 collisions are required on the average to thermalize a 2 mev neutron in graphite.

The *slowing down power* of a material is defined as the product, $\xi\Sigma_s$. This product is more significant for purposes of comparison since it reflects not only the average energy change per scattering collision, but also the probability (Σ_s) that such collisions will occur. In order to include the likelihood of absorption during slowing down, the *moderating ratio* is defined as $\xi\,\Sigma_s/\Sigma_a$. Comparison of some moderators is given below:

Moderator	$\xi\Sigma_s$, cm^{-1}	$\xi\Sigma_s/\Sigma_a$
water	1.53	70
heavy water (99.75% pure)	0.174	5,800
carbon	0.063	170

Slowing Down Density

The number of neutrons per cu cm per sec which are slowing down past a given energy level E is known as the *slowing down density*, q. The relation between q and previously established quantities will now be developed, and it will be found that it is possible to evaluate the resonance escape probability from this analysis of slowing down density. Furthermore, this concept is necessary to the later development of the Fermi age equation.

Suppose q_o fast neutrons of energy E_o are being produced in a medium each second. Let E_1 be an energy level between E_o and thermal energy, and let ΔE_1 be a small interval about E_1. Let E be an arbitrary energy such that neutrons might be scattered from E to ΔE_1, and let dE be an infinitesimal interval about E. Neutrons might be scattered into ΔE_1 from as high an energy as E_1/α and might be scattered from E to as low a level as αE, so E_1/α and αE represent the limits of energy level for a single scattering process from E or into E_1. Figure 9 shows these energy bands in a schematic way.

Let there be n dE neutrons per cu cm having energies between E and E+dE (note this is not the usual definition of n), and observe that, at steady state, the rate at which neutrons enter the interval ΔE_1 must equal the leaving rate. The leaving rate is just the flux times the scattering cross section, $(n_1\,\Delta E_1)\,v_1\,\Sigma_s$. The entering rate may be found by integrating from E_1 to E_1/α the rate of scatter into ΔE_1 from dE, which is

$$\frac{\Delta E_1}{E-\alpha E}(n\,dE)\,v\,\Sigma_s$$

since the probability of scatter from E to ΔE_1 is just the ratio of the energy interval, ΔE_1, divided by the total possible energy interval, E—αE, and the total number scattered out of dE is (n dE) v Σ_s, the product of neutron flux times the cross section.

Equating the rate of entry of neutrons into ΔE_1 to the leaving rate gives

$$nv_1\,\Sigma_s\,\Delta E_1=\int_{E_1}^{E_1/\alpha}\frac{\Delta E_1}{E-\alpha E}nv\,\Sigma_s\,dE$$

This equation may be solved to give

$$nv\,\Sigma_s=c/E$$

where c is a constant. (The method of solution will not be given here. The slowing down density at E_1 is, by definition, the rate at which neutrons are slowing down past this energy level. We may write

$$q_1=\int_{E_1}^{E_1/\alpha}nv\,\Sigma_s\,dE\,\frac{E_1-\alpha E}{E-\alpha E}$$

since the ratio $(E_1-\alpha E)/(E-\alpha E)$ is the probability that a neutron will slow down past E_1 after a single collision at an energy higher than E_1. Substituting $n\,v\,\Sigma_s=c/E$ yields

$$q_1=c\left(1+\frac{\alpha}{1-\alpha}\ln\alpha\right)=c\xi$$

$$\text{or } c=q_1/\xi$$

But E_1 is an arbitrary energy level, and it follows that

$$c=q/\xi \qquad \text{or } nv=q/\xi\,\Sigma_s\,E \qquad [7]$$

Consideration of Figure 9 shows the preceding derivation would not apply when the source energy, E_o, was approached too closely. For E_o=2 mev, the above relationship between flux and slowing down density is valid for E<0.7 mev with carbon and E<0.003 mev for deuterium. It is valid at all energies for hydrogen.

Slowing Down with Capture

It is possible to find the slowing down density rather easily in the special case of a homogeneous mixture of a moderator and a heavy neutron ab-

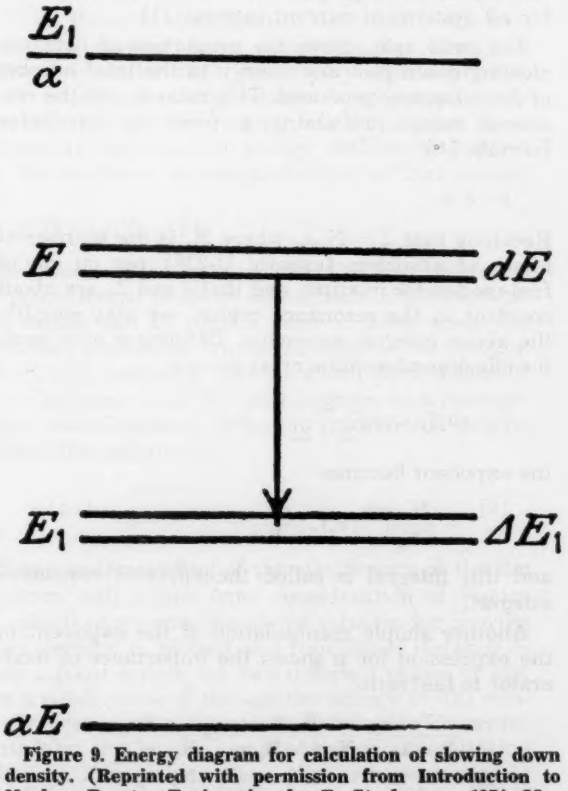


Figure 9. Energy diagram for calculation of slowing down density. (Reprinted with permission from Introduction to Nuclear Reactor Engineering by R. Stephenson, 1954, McGraw-Hill Book Company.)

sorber such as uranium. If we recall that absorption takes place, not uniformly at all energies between E_o and thermal energy, but only over certain fairly narrow energy bands (resonance peaks), it is possible to write for the resonance region

$$nv=\frac{q}{\xi(\Sigma_s+\Sigma_a)E}$$

This is analogous to the expression for neutron flux without absorption

$$nv=q/\xi\Sigma_s E$$

which was derived above.

In a resonance region, furthermore, the absorption of neutrons will cause a decrease in q, i.e.,

$$dq=nv\,\Sigma_a\,dE$$

Combining the above expressions and integrating

$$\int_q^{q_0}\frac{dq}{q}=\int_E^{E_0}\frac{\Sigma_a}{\xi(\Sigma_a+\Sigma_s),}\frac{dE}{E}$$

$$\frac{q}{q_0}=e^{-\int_E^{E_0}\Sigma_a\,dE/\xi(\Sigma_a+\Sigma_s)E}$$

This equation is strictly applicable only to a homogeneous mixture of hydrogen and a heavy absorber such as U-238. However, it gives a good approximation for uranium-graphite and a fair approximation for all systems of current interest (1).

The ratio, q/q_o, gives the proportion of neutrons slowing down past any energy to the total number of fast neutrons produced. This ratio is just the resonance escape probability, p, from the four-factor formula [1].

$$p=q/q_0$$

Recalling that $\Sigma_a=N_o\sigma_a$, where N_o is the number of atoms of absorber (usually U-238) per cu cm of fuel-moderator mixture, and that ξ and Σ_s are about constant in the resonance region, we may simplify the above integral somewhat. Defining a new term, the effective *absorption cross section,*

$$(\sigma_a)_{eff}=\sigma_a\frac{\Sigma_s}{\Sigma_s+\Sigma_a}$$

the exponent becomes

$$-\frac{N_o}{\xi\Sigma_s}\int_E^{E_0}(\sigma_a)_{eff}\frac{dE}{E}$$

and this integral is called the *effective resonance integral.*

Another simple manipulation of the exponent in the expression for p shows the importance of moderator to fuel ratio:

$$\frac{\Sigma_a}{\Sigma_s+\Sigma_a}=\frac{N_o\sigma_a}{N_m\sigma_s+N_o\sigma_a}=\frac{1}{\frac{N_m}{N_o}\frac{\sigma_s}{\sigma_a}+1}$$

Since the microscopic cross sections are fixed by the materials, there exists a factor in the integrand which becomes smaller as N_m/N_o becomes greater, and p will approach unity as the relative amount of moderator in the core becomes large.

FERMI AGE THEORY

We have discussed the mechanism by which neutrons are slowed to thermal energy and the way that neutrons of constant energy diffuse through a non-absorbing medium. Another important segment of the theory which is needed is the analysis of spatial distribution of neutrons of various energies during slowing down. The average "crow-flight" or vector distance that a neutron travels from the point of its production as a fast neutron to the point at which it has only thermal energy is important in determining the neutron leakage rate in a reactor.

Continuous Slowing Down

An approximation which is useful to an analysis of the slowing down process is that, since a neutron experiences a large number of collisions with moderator nuclei, it may be assumed that each collision results in about the same change in the logarithm of the neutron energy. Figure 10 shows the stepped curve of ln E versus time which would result if each collision resulted in the change ξ in ln E, where ξ by definition is the *average* change in that quantity during slowing down. In this curve, the vertical steps are a distance ξ apart, but the horizontal portions become longer as the velocity is reduced, since a greater time elapses between collisions.

For nuclei of moderate or high mass number, ξ is small and a large number of collisions are required to thermalize a fast neutron. In this case, it is permissible to replace the actual slowing down curve with the *continuous slowing down model,* which is the curve of continuous slope in Figure 10. For a hydrogenous moderator, it will be recalled that it is possible for a fast neutron to lose all its energy in one collision, so the continuous slowing down model is not adequate. Even for heavy water, however, it has been found that the continuous slowing down model gives reasonably good results. A more com-

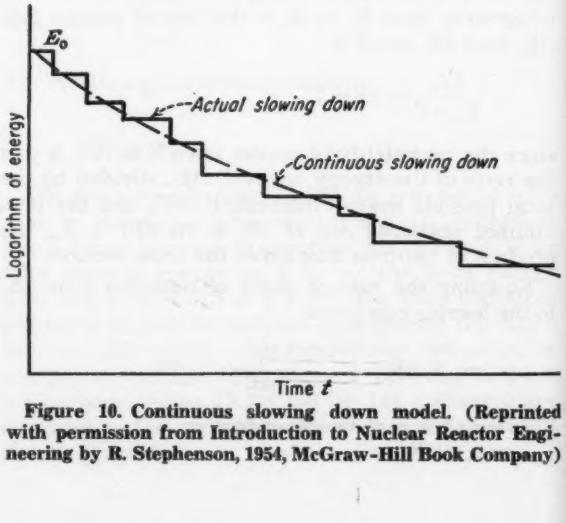


Figure 10. Continuous slowing down model. (Reprinted with permission from Introduction to Nuclear Reactor Engineering by R. Stephenson, 1954, McGraw-Hill Book Company)

plicated extension of the theory is available for hydrogenous moderators (1).

The Age Equation

It is possible to develop an approximate analysis, based on the continuous slowing down model, which will relate the slowing down density, q, to the neutron energy during slowing down and to properties of the moderator. Consider a neutron at any energy, E, between fast neutron energy, E_o, and thermal energy, E_{th}. The energy E will be attained by the neutron at a time, t, later than the neutron's production from the fission of a fuel nucleus. If v is the velocity of the neutron at time t, then it will travel a distance v dt and will probably have $v\,dt/\lambda_s$ scattering collisions during the infinitesimal time interval, dt. Since the average change in ln E per scattering collision is ξ, by definition, we may write for the *decrease* in ln E with time,

$$-\frac{\partial(\ln E)}{\partial t}\,dt=\frac{\xi\,v\,dt}{\lambda_s}$$

Recall that

$$E=\tfrac{1}{2}\,mv^2 \quad \text{or } v=\sqrt{2E/m}$$

$$\text{and } \frac{\partial(\ln E)}{\partial t}=\frac{1}{E}\frac{\partial E}{\partial t}$$

$$\text{so that } \frac{\partial E}{\partial t}=-\frac{\xi E^{3/2}}{\lambda_s}\sqrt{\frac{2}{m}}=-\frac{\xi E v}{\lambda_s}$$

$$\text{or } \frac{\partial t}{\partial E}=-\frac{\lambda_s}{\xi E v}$$

It has been shown previously that the slowing down density, q, without absorption or leakage, is

$$q=\xi\,\Sigma_s\,\Phi\,E$$

$$\text{or } \nabla^2 q=\xi\,\Sigma_s\,E\,\nabla^2\Phi$$

since the flux is the only term in the equation for q which is a function of the space coordinates. (Note that the Laplacian operator ∇^2 implies partial differentiation with respect to the space coordinates but not with respect to time). But the basic equation [4a] which expresses the time rate of increase of the neutron population, $\partial n/\partial t$, is just equal to the diffusion term when there is no production or absorption, i.e.,

$$\frac{\partial n}{\partial t}=D\,\nabla^2\Phi$$

For neutrons of constant velocity, since $\Phi=n\,v$ and

$$\frac{\partial q}{\partial t}=\xi\,\Sigma_s\,E\frac{\partial\Phi}{\partial t},\quad \frac{\partial n}{\partial t}=\frac{1}{v}\frac{\partial\Phi}{\partial t}=\frac{1}{v\,\xi\,\Sigma_s\,E}\frac{\partial q}{\partial t}$$

Hence,

$$\frac{\partial q}{\partial t}=v\,\xi\,\Sigma_s\,E\left(\frac{\partial n}{\partial t}\right)=(v\,\xi\,\Sigma_s\,E)\,(D\,\nabla^2\Phi)=D\,v\,\nabla^2 q$$

Combining,

$$\frac{\partial q}{\partial E}=\frac{\partial q}{\partial t}\frac{\partial t}{\partial E}=(D\,v\,\nabla^2 q)\left(-\frac{\lambda_s}{\xi E v}\right)=-\frac{D}{\Sigma_s\,\xi\,E}\nabla^2 q$$

The symbol τ is used for *Fermi age* and is defined by the equation

$$d\tau=-\frac{D\,dE}{\Sigma_s\,\xi\,E}$$

Transforming the independent variable in the partial differential equation for q gives the *Fermi age equation*

$$\nabla^2 q=\frac{\partial q}{\partial \tau} \qquad [8]$$

It is seen that this equation has the same mathematical form as the heat conduction equation, which may be written as

$$\nabla^2 T=\frac{1}{\alpha}\frac{\partial T}{\partial t}$$

where T is temperature, t is time and α is a property of the medium called thermal diffusivity. Because of the similarity between τ and t in these equations, τ is called the age of the neutrons. It should be observed, however, that τ has the units of square centimeters, not of seconds. Nevertheless, Fermi age is related to the time of existence of the neutrons from the instant of formation until the instant that an energy E is attained, and τ grows steadily larger as the energy decreases. It is arbitrarily stated that age is zero at the instant of formation of a fast neutron, i.e., $\tau(E_o)=0$.

When there is absorption of neutrons in the medium, the slowing down density at any energy is naturally smaller than age theory would predict. For a weakly absorbing medium, a good approximation may be found by multiplying the slowing down density at any neutron energy without absorption by the resonance escape probability at that energy, i.e.,

$$q^*(E)=p(E)\,q(E)$$

Solutions of the age equation [8] may be found by conventional methods when boundary conditions are simple and by approximate methods when the boundary conditions are not so simple. One solution which is of interest is that for a unit point source of fast neutrons, all of the same energy, in a homogeneous non-absorbing diffusion medium of infinite extent. This solution is

$$q(r,\tau)=\frac{e^{-r^2/4\tau}}{(4\pi\tau)^{3/2}} \qquad [9]$$

Some understanding of the significance of the age equation will follow from consideration of Figure 11, which represents the above solution for slowing down density, q, plotted as a function of distance, r, from a point source, for two different values of age. For a small value of the age the energy is still relatively great and the spatial distribution is represented by the higher curve. After more scattering collisions have taken place, the average energy is smaller and τ is larger. The greater number of collisions means the neutrons have diffused farther

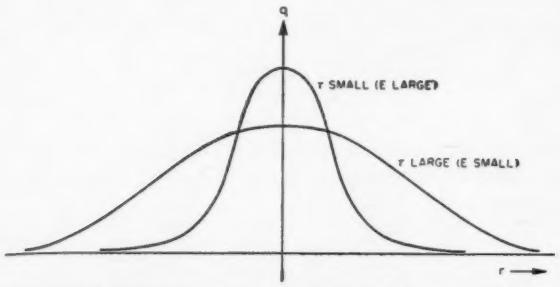


Figure 11. Typical curves of slowing down density distribution about a point source. (Reprinted with permission from Principles of Nuclear Reactor Engineering by S. Glasstone, 1955, Van Nostrand Company)

from the source and the low, flat curve of slowing down density is obtained.

Slowing Down Length

A more precise physical interpretation may be attached to the age by means of a calculation similar to that performed previously for the diffusion length. It will be recalled that we calculated the mean square distance travelled by a thermal neutron before it was captured and found this to be

$$\overline{r^2}=6\,L^2$$

Let us perform a similar calculation to find the mean square distance a neutron travels while its energy decreases from E_0 to E_{th}. The defining equation for $\overline{r_s^2}$ for a point source in an infinite medium is

$$\overline{r_s^2}\int_0^\infty 4\pi\, r^2 q(r,\tau)\,dr=\int_0^\infty r^2\left[4\pi\, r^2\, q(r,\tau)\right]dr$$

Substituting the age equation solution [9] given above for q with this geometry,

$$\overline{r_s^2}=\frac{\int_0^\infty r^4\, e^{-r^2/4\tau}\,dr}{\int_0^\infty r\, e^{-r^2/4\tau}\,dr}$$

Using integral tables,

$$\overline{r_s^2}=6\,\tau \quad \text{or} \quad \tau=\overline{r_s^2}/6$$

Analogously to diffusion length, the *slowing down length* is defined in terms of age as $\sqrt{\tau}$. A *migration area*, M^2, may be defined as

$$M^2=L^2+\tau$$

and a *migration length*, M, as

$$M=\sqrt{L^2+\tau}$$

Values of diffusion length, slowing down length and migration length are given below for the common moderators:

Moderator	L (cm)	$\sqrt{\tau}$ (cm)	M (cm)
water	2.88	5.74	6.43
heavy water	100	10.9	101
beryllium	23.6	9.90	25.8
graphite	50.0	18.7	53.6

The migration length is a measure of the average "crow-flight" distance travelled by a neutron during slowing down and during thermal diffusion, i.e., the average distance from the point of production of the fast neutron to the point of absorption of the same neutron after thermalization and diffusion.

THE CRITICAL EQUATION

We have now developed equations for the source, diffusion and absorption terms in the neutron diffusion equation, which becomes, for steady state,

$$D\,\nabla^2\Phi(r)-\Sigma_a\,\Phi(r)+p\,q(\tau_{th},r)=0 \qquad (4b)$$

In this equation, the variable r refers to whichever space coordinates are chosen, and the source term, S, has been replaced by the product of resonance escape probability and slowing down density, which it was previously stated would apply to a weakly absorbing medium. If we make our balance on thermal neutrons, q varies with position only, as the equation indicates, since τ_{th} is a property of the medium.

In order to find $q(\tau_{th},r)$, we must solve the age equation

$$\nabla^2 q(\tau,r)=\frac{\partial q(\tau,r)}{\partial\tau} \qquad (8)$$

As a boundary condition, recall that the age is zero when neutron energy is maximum, i.e., for fission neutrons. Hence q(0, r) is the number of neutrons of fission energy per cubic centimeter per second.

Another expression for this number of fast neutrons may be found by considering that the rate of thermal neutron capture in all processes is $\Phi\,\Sigma_a$. The fraction of $\Phi\,\Sigma_a$ captured in the fuel is f, the thermal utilization factor, and each thermal neutron capture in fuel results in the production of $\eta\epsilon$ fast neutrons. Hence, the number of neutrons of fission energy may be written as $\Phi\,\Sigma_a\,f\,\eta\epsilon$, or

$$q(0,r)=\Phi\,\Sigma_a\,f\,\eta\,\varepsilon=\Phi\,\Sigma\,k/p \qquad [10]$$

since $k=\eta\epsilon\,p\,f$. We now have a relation between q and Φ which relates the diffusion equation and the age equation.

Let us solve the age equation by the method of separation of variables. Assume

$$q(\tau,r)=T(\tau)\,R(r)$$

Substituting into the age equation gives

$$\frac{\nabla^2 R(r)}{R(r)}=\frac{1}{T(\tau)}\cdot\frac{dT(\tau)}{d\tau}$$

Since r and τ are independent, we may set each side equal to a constant, $-B^2$, called the *buckling* of the system.

$$\frac{\nabla^2 R}{R}=-B^2 \quad \text{or} \quad \nabla^2 R+B^2 R=0$$

$$\frac{1}{T}\frac{dT}{d\tau}=-B^2 \quad \text{or} \quad T=Ae^{-B^2\tau}$$

where A is a constant. Using this solution for $T(\tau)$, the slowing down density is

$$q(\tau,r)=T(\tau)\,R(r)=A\,R\,e^{-B^2\tau}$$

When $\tau=0$, we may use the expression [10] for $q(0,r)$

$$q(0,r)=\Phi\,\Sigma_a\,k/p=A\,R$$

from which

$$\nabla^2 R=\frac{\Sigma_a\,k}{A\,p}\nabla^2\Phi$$

and, since $\nabla^2R+B^2R=0$, we obtain the important equation

$$\nabla^2\Phi+B_g^2\,\Phi=0 \qquad [11]$$

The constant B^2 is given the subscript to denote *geometrical buckling*. Since the flux in this equation is a function of position only, the value of B_g^2 which satisfies this equation depends only on the size and shape of the reactor.

Another important relation which can be obtained from the above equations is called the *critical equation* and is

$$\frac{k\,e^{-B_m^2\tau_{th}}}{1+L^2\,B_m^2}=1 \qquad [12]$$

The condition for criticality is that B^2 satisfy both of the preceding two equations, or that B_m^2 and B_g^2 are identically equal. However, it is convenient to distinguish between them, since, as was just pointed out, B_g^2 must satisfy geometrical limitations, and, in the second equation, B_m^2 is determined by the reactor materials. This property, the *material buckling*, varies with k, τ_{th} and L, all of which depend on the fuel and moderator properties and on their arrangement, but not on actual size or shape of the core.

Critical Size of a Reactor Core

We can now solve the differential equation in which the buckling appears. For a spherical reactor core, this equation becomes

$$\frac{d^2\Phi}{dr^2}+\frac{2}{r}\frac{d\Phi}{dr}+B^2\,\Phi=0$$

which has the general solution

$$\Phi(r)=\frac{A\sin Br}{r}+\frac{C\cos Br}{r}$$

The boundary conditions require that the flux be finite when r=0, so the arbitrary constant, C, must be zero. The other boundary condition is that the fluz be zero at a radius R, which requires that

$$\frac{A}{R}\sin BR=0$$

or $BR=n\pi$ and $B^2=\left(\frac{n\pi}{R}\right)^2$

where n is an integer. Further consideration shows that the only integer which will give positive values of Φ throughout the range of r is unity. Hence

$$B^2=\left(\frac{\pi}{R}\right)^2$$

$$\Phi(r)=\frac{A}{r}\sin\frac{\pi r}{R}$$

where the constant A is determined by the power level at which the reactor is operating.

Figure 12 shows the three common geometrical arrangements with their appropriate systems of space coordinates. The following table, taken from Glasstone (1), summarizes the solutions for buckling and flux distribution from these three geometries:

Geometry	Buckling	Critical Flux Distribution	Minimum Critical Volume
Sphere	$\left(\frac{\pi}{R}\right)^2$	$\frac{A}{r}\sin\frac{\pi r}{R}$	$\frac{130}{B^3}$
Rectangular parallelepiped	$\left(\frac{\pi}{a}\right)^2+\left(\frac{\pi}{b}\right)^2+\left(\frac{\pi}{c}\right)^2$	$A\cos\frac{\pi x}{a}\cos\frac{\pi y}{b}\cos\frac{\pi z}{c}$	$\frac{161}{B^3}$
Finite cylinder	$\left(\frac{2.405}{R}\right)^2+\left(\frac{\pi}{H}\right)^2$	$A\,J_o\left(\frac{2.405\,r}{R}\right)\cos\frac{\pi z}{H}$	$\frac{148}{B^3}$

Figure 13 shows the variation of the sine, cosine and Bessel functions which represent the solutions to the flux equation.

If the shape and size of the reactor core are specified, then, the geometrical buckling can be found as shown above. For criticality, the material buckling must be the same, so this value of B^2 may be substituted into the critical equation. The composition and

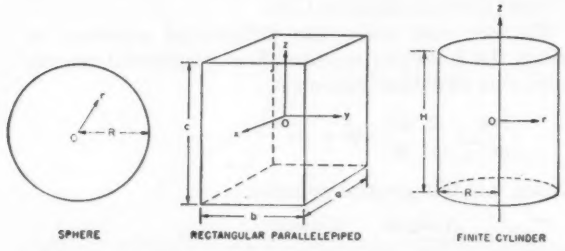


Figure 12. Dimensions and coordinates for reactors of three geometries. (Reprinted with permission from Principles of Nuclear Reactor Engineering by S. Glasstone, 1955, Van Nostrand Company)

arrangement of the fuel may then be adjusted to satisfy that equation, which relates k, L^2 and τ to B^2.

Further consideration of the critical equation

$$\frac{k\, e^{-B^2\tau_{th}}}{1+L^2B^2}=1$$

reveals that the term $e^{-B^2\tau_{th}}/(1+L^2B^2)$ must represent a correction factor for neutron leakage from the core, since k=1 for a critical reactor of infinite extent (i.e., no leakage). Hence this factor is just the non-leakage probability, P, which was discussed earlier in connection with the definition [2] of k and k_{eff}:

$$P=\frac{e^{-B^2\tau_{th}}}{1+L^2B^2}$$

SUMMARY

This brief introduction to reactor design theory is presented to introduce the subject to technically competent persons in other fields. Of necessity, certain portions of the discussion have become mathematically involved, but it is hoped that the description and conclusions will be meaningful to most readers in spite of this difficulty. A more serious problem which is encountered in attempting to present a brief description of the methods of reactor analysis is that of superficiality and lack of rigor. It is assumed that the reader who finds the scope of this material too limited will refer to the excellent textbooks which are now available in this field (1, 2, 4, 6).

To summarize the reactor analysis which has been described, let us suppose we wish to calculate the critical size of a graphite moderated reactor using pure U-235 as fuel. The first step would be to evaluate the terms in the four-factor formula to find k

$$k=\eta\,\varepsilon\,p f$$

For an enriched system such as this, the moderator to fuel ratio will be high and the resonance escape probability, p, will be close to unity. As a first approximation, assume the product, ϵ p, is equal to one. Furthermore, it is known that the average number of neutrons produced per fission in U-235 is 2.1, so

$$k=2.1\,f \qquad (3)$$

The thermal utilization factor depends on the fuel arrangement in the moderator and on the relative amounts of moderator and fuel in the core. If we assume a homogeneous arrangement, it is easy to show that

$$f=\frac{\Sigma_{au}}{\Sigma_{au}+\Sigma_{am}} \qquad (13)$$

if we ignore absorption by impurities. For a composition of 10,000 atoms of carbon to one of U-235, f is 0.88 and k is 1.85.

It remains to evaluate the diffusion length L in the critical equation [12] before we can solve for the buckling. Recalling the definition of diffusion length

$$L^2=\frac{D}{\Sigma_a}$$

it will be seen that for the very high moderator to fuel ratio which is permitted by the use of U-235, the diffusion coefficient D for the mixture is practically the same as that for the moderator alone. Absorption in the fuel cannot be ignored, however, so

$$L^2=\frac{D_m}{\Sigma_{am}+\Sigma_{au}} \qquad [14]$$

A simple algebraic manipulation between [13] and [14] yields

$$L^2=L_m^2(1-f)$$

where L_m^2 refers to the pure moderator. If this expression and [3] above are substituted into the critical equation [12] we obtain

$$2.1\,f\,e^{-B^2\tau_{th}}=1+L_m^2(1-f)B^2 \qquad [15]$$

Since τ_{th}=350 sq. cm and L_m=50.0 cm for graphite, we may solve [15] for the buckling, which proves to be 0.00136 $(cm)^{-2}$. This constant may then be used

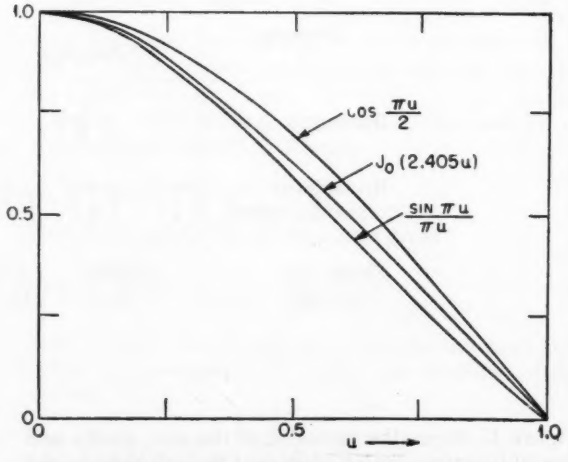


Figure 13. Functions for determination of neutron flux distribution. (Reprinted with permission from Principles of Nuclear Reactor Engineering by S. Glasstone, 1955, Van Nostrand Company)

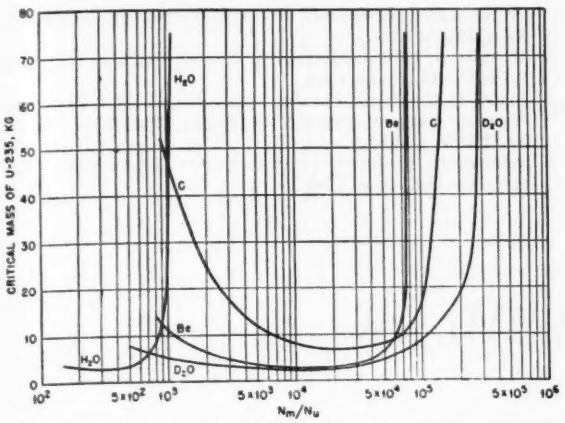


Figure 14. Relationship between critical mass and composition of bare, spherical reactors with pure U-235 as fuel. (Reprinted with permission from Principles of Nuclear Reactor Engineering by S. Glasstone, 1955, Van Nostrand Company)

with equation [11] to find the minimum critical dimensions of the reactor core. For a spherical reactor, for example, the radius would be about 85 cm and the critical mass would be about eight kilograms of U-235. (While these calculations are useful as a first approximation, a neutron reflector would surely be provided around the core and the actual critical radius and critical mass would be smaller than the above values.) Figure 14 shows the variation of critical mass with composition ratio for bare, spherical reactors with pure U-235 as fuel. Results are included for all the common moderators.

NOMENCLATURE

A	Atomic weight of an element. Also used for mass number of a nucleus.
B^2	Buckling, cm^{-2}.
D	Diffusion coefficient, cm, defined as $\lambda_{tr}/3$.
E	Energy, mev. Neutron kinetic energy varies from E_0 for a fission neutron down to E_{th} for a thermal neutron.
f	Thermal utilization factor, defined as the ratio of thermal neutrons absorbed by fuel to thermal neutrons absorbed by all processes.
J	Neutron current density, neutrons/(sq cm) (sec), defined in terms of a particular direction.
J_0	Bessel function of zero order.
K^2	A constant, cm^{-2}, defined as Σ_a/D.
k	Multiplication factor, defined as the ratio of the number of neutrons in any one generation to the number in the preceding generation.
L	Diffusion length, cm, related directly to the mean distance travelled by a thermal neutron from source to point of absorption.
M	Migration length, cm, related directly to mean distance travelled by a neutron from point of birth as a fast neutron to point of absorption as a slow neutron.
m	Mass of a particle, grams. Also, a subscript referring to moderator.
N	Number of atoms of any substance in a cu cm of mixture.
n	Number of neutrons in a cu cm of a volume being studied, except in *Slowing Down Density* where n dE is this number. Also, n is used to denote an integer.
P	Nonleakage probability, a correction to the multiplication factor to account for neutron leakage from the surface of a reactor.
p	Resonance escape probability, defined as the fraction of neutrons which reach thermal energy without capture.
q	Slowing down density, neutrons/(cu cm) (sec), slowing down past a given energy level, E.
q^*	Slowing down density, neutrons/(cu cm) (sec), in a weakly absorbing medium.
Q	Strength of a neutron source, neutrons/(sec) (sq cm).
R	Radius, cm.
r	Length variable, cm.
$\overline{r^2}$	Mean square distance travelled by a neutron during diffusion.
$\overline{r_s^2}$	Mean square distance travelled by a neutron during slowing down.
S	Source term in neutron balance equation, neutrons/(cu cm) (sec).
t	Time, sec.
u	Subscript referring to fuel, including both U-238 and U-235 or other fissionable isotope.
v	Velocity, cm/sec.
α	Maximum energy loss ratio in elastic collision of a neutron with a nucleus, $\left(\frac{A-1}{A+1}\right)^2$
ε	Fast fission factor, defined as the ratio of the number of all fast neutrons produced to the number produced by thermal neutron fission.
η	Number of fast neutrons produced per slow neutron absorbed in the fuel.
θ	Scattering angle in the center of mass (C) system of coordinates. Also used to denote angular measure in general.
λ	Mean free path, cm. Various subscripts are used (see under Σ).
ν	Average number of fast neutrons produced per fission.
ξ	Average energy decrement or natural logarithm of the energy change per collision.
Σ	Macroscopic cross section, cm^{-1}. Various subscripts are used as follows: a absorption; f fission; s scattering; tr transport
σ	Microscopic cross section, barns. Various subscripts are used (see under Σ).
$(\sigma_a)_{eff}$	Effective absorption cross section, barns.
τ	Fermi age, sq cm.
Φ	Neutron flux, nv, neutrons/(sq cm) (sec).
ϕ	Angular measure.
Ψ	Scattering angle in the laboratory (L) system of coordinates.

REFERENCES

1. Glasstone, S., *Principles of Nuclear Reactor Engineering.* 1955. Van Nostrand, New York. This is a recent textbook which gives an authoritative presentation of the most im-

portant aspects of the subject. The chapters on reactor theory are more than just an abridgement of reference 2; new developments and additional practical design factors have been included. This text was used for many of the examples and data in the present article.

2. Glasstone, S., and Edlund, M. C., *Elements of Nuclear Reactor Theory*. 1952. Van Nostrand, New York.
3. Leppert, G., "An Introduction to Reactor Physics," JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Volume 67, No. 4, 1955, p. 855.
4. Murray, R. L., *Introduction to Nuclear Engineering*. 1954. Prentice-Hall, New York.
5. Pigott, E. C., Murex Review, Vol. 1, No. 14, 1954. Reprinted JOURNAL AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. 67, No. 2, 1955, p. 375.
6. Stephenson, R., *Introduction to Nuclear Engineering*. 1954. McGraw-Hill, New York.

TECHNICAL PROGRESS IN SHIPBUILDING DURING 1955.

ACKNOWLEDGEMENT

This staff article was published in the January, 1956, edition of "The Shipbuilder and Marine Engine-Builder."

In reviewing the technical progress in shipbuilding during another year, it is fitting to make some comments on the general position of shipbuilding in this country and in others throughout the world. Our source for this information is the statistical returns of Lloyd's Register of Shipping; and due to the fact that, at the time of writing, the figures for the year cannot be complete, we are obliged to base our remarks on the latest available statistics, *viz.*, those up to the end of the third quarter of 1955.

On the basis of tonnage completed during the quarter, Great Britain and Northern Ireland are still in the lead, with a total of 304,002 tons gross, out of a total of 1,193,593 tons, or 71 ships out of a total of 337. Britain's most serious competitor is Germany, with 95 ships of 241,538 tons completed for the quarter. The next important challengers are Japan, with 46 ships of 205,022 tons, and Sweden, with 17 ships of 101,302 tons. Only six ships, of 45,212 tons, were completed in the United States of America during the quarter.

If these figures are compared with those for Great Britain and Germany for the corresponding quarter of 1954, there appears to have been a decline in the tonnage produced in both countries.

It is instructive, also, to compare the figures for ships under construction. At the end of the third quarter of 1955, Great Britain had under construction 346 ships of 2,147,057 tons, out of a world total of 1,349 ships of 6,285,699 tons. Germany had under construction 256 ships of 713,838 tons. Considering these figures in relation to those for ships completed, while Britain has, by far, the greatest amount of tonnage under construction, the production rate in Germany (and in some other countries) is higher than here. This is a matter which is, of course, causing great concern to British shipowners and shipbuilders alike, as reduced production rate almost certainly means higher first costs. While, therefore, there seems to be no cessation in the demand for tonnage, and the shipyards of this country have still plenty of work in hand, there is no room for complacency, and clearly every effort should be made to reduce building costs.

So far as numbers of ships being built in this country are concerned, the cargo liner occupies first place; there were 100 of these vessels under construction at the time of writing. From the tonnage point of view, however, the oil-tank vessel still occupies first place, there being 79 ships, of nearly 1,000,000 tons gross, in hand at present.

DESIGN OF SHIPS

In this section, brief reference will be made to various technical aspects of design which have been discussed in technical literature during the past year, rather than the consideration of the design of particular vessels.

Sir William Currie, G.B.E., had much excellent information to give on the subject of design of ships in his paper entitled "Liners of the Past, Present and Future on Service East of Suez," which he read at the last spring meetings of the Institution of Naval Architects.

The comparison with ships of the past and the present in this paper, was made between the *Marmora* (built in 1903) and the *Arcadia* (built in 1954). On the basis of general characteristics, the newer ship showed very much larger dimensions and displacement, as well as greater speed and power, than the older vessel.

Evidence of the advances in marine engineering achieved in this period is obtained by comparing the machinery of the two ships. While the older ship had twin quadruple-expansion engines, with steam supplied by coal-fired boilers, the new ship had twin Parsons-type geared turbines, with oil-fired water-tube boilers.

The vast change in the type of machinery had a very marked effect on the weight required. Although the power available in the new ship was more than three times as great as that in the 1903 ship, the corresponding increase in machinery weight was under 50 per cent. Features of the modern ship which are different from the older one are the introduction of welding into the construction of the hull; the great improvement in life-saving appliances which has taken place in the period; and vast changes in the standards of accommodation for both crew and passengers.

With regard to life-saving, it is worthy of note that, while both ships had the same number of lifeboats, those of the present-day ship are capable of accommodating over 2,000 persons, as against under 1,000 in the old ship, which indicates the growth in size of the lifeboat in the period.

While the steel weight, as a fraction of the displacement, has decreased from 0.396 to 0.351—indicating improvements in ship structures—the hull weight has increased from 0.529 to 0.575. These last-mentioned figures are an indication of the growth in the amenities provided for passengers.

With regard to the ships of the future, Sir William considered that much depends upon the naval architect in producing more economical ship forms, which will reduce fuel, and hence running costs. He sees, too, that high first costs are likely to prove a stumbling block to the replacement of obsolescent tonnage by modern ships capable of competing in world trade.

The question of the design of more efficient ships—this time, of the tramp type—was the subject of a paper entitled "Design of Economic Tramp Ships," by Mr. E. C. B. Corlett, M.A., Ph.D., and which was read before the Institution of Naval Architects in September, 1955. In this contribution, the author showed that the majority of ships of the type being built today are of about 10,500 tons deadweight and 15 knots speed. It is demonstrated that this is about the economical speed for this class of ship.

Mr. Corlett's paper contained much that is of interest to the designer. For instance, machinery aft is advocated, together with a combined system of longitudinal framing in the deck and bottom, and transverse framing in the sides. It is also recommended that the girders should be fitted above deck, so as to free the 'tween decks of obstructions. Modern cargo-handling appliances, in the form of cranes, are suggested, to replace the traditional winches and derricks, and thus improve the turn-round time—a feature which causes great anxiety to shipowners.

A novel feature of the proposed designs for modern ships, is the adoption of a form which would enable the shell plating to be developed, and thus save much in construction costs. This type of construction would involve the adoption of a straight-line type of sections in some parts of the form, and it is shown that this would not necessarily result in adverse effects from the propulsion point of view.

It is clear, from a paper such as this, that there are many ways in which the first cost of a ship may be reduced, and also that the vessel may be improved as a freight-earning unit by a careful consideration of all the factors involved in its design.

Much useful information on a specialized type of ship is to be found in a paper entitled "Modern Ore-carriers," read by Mr. J. J. Henry before the American Society of Naval Architects and Marine Engineers in May last. The author of this valuable paper deals comprehensively with all aspects of the design of these vessels, and includes not only the iron ore-carrier, but also the bauxite-carrier, as well as ships designed to carry other dry cargoes, or petroleum, in addition to ore.

Technical features which influence design are discussed, and also such matters as bilge and ballast pumping arrangements, steering gears, and anchor and mooring arrangements. The design of hatch covers is gone into in some detail, and the impact of classification society requirements on design is dealt with.

Not the least valuable information in this paper are the full data concerning not only ocean-going and Great Lakes ore-carriers, but bauxite-carriers as well. These data include form particulars, weight of steel, outfit and machinery, and many details of fittings.

Finally, in this section dealing with the general design of ships, mention should be made of a paper entitled "An Aluminum River Tanker Design," by Mr. E. C. B. Corlett, M.A., Ph.D., and Mr. K. J. Fryer, which was read before the Institution of Engineers and Shipbuilders in Scotland in November last. The design is given in the form of a comparison with a steel ship.

Normally, if a deadweight-carrier, such as a tanker, is designed in aluminum alloy, it will usually be found that, on the basis of equivalent strength, the weight saved, although considerable, is not sufficient to justify the additional expenditure on an aluminum-alloy ship.

In the present instance, however, the author has shown that, on the basis of the standard longitudinal-strength calculation, the stress obtained in the all-steel vessel is very low, of the order of 1.8 ton per sq. in. It was concluded, therefore, that the scantlings of the steel vessel, as determined from classification society rules, were heavier than need be from purely strength considerations, and that the increases were corrosion allowances, because of the heavy wastage due to corrosion in the oil tanks of such vessels.

It was further considered that these heavy allowances would not need to be made in the aluminum-alloy design, because of the greater inherent resistance to corrosion of this material. On this basis, the scantlings of side shell were increased by only 15 per cent over those of the steel ship, and the deck and the bottom were increased by 20 per cent.

The net result was that, while the steel ship carried 320 tons of cargo, this could be increased by 25 per cent to give 400 tons of cargo in the aluminum ship. It was finally estimated that the aluminum ship would cost £12,500 more, but that the first cost per ton of the

additional cargo was less than the first cost per ton of cargo in the steel ship. The comparison is an interesting one, which is worthy of very careful study.

STRUCTURAL STRENGTH AND WELDING

The relation between the classification societies and the development of structures of ships is an important one, and we cannot do better, at the commencement of this section on structural strength, than to refer to a paper by Mr. J. M. Murray, M.B.E., B.Sc., entitled "A Hundred Years of Lloyd's Register Rules," and which was read at the last spring meetings of the Institution of Naval Architects. This paper recounted the growth of, and modification to, the Rules from 1855. During this period, a major change in the material of construction of ships has taken place, *i.e.*, iron first replaced wood, and, at a later stage, it was itself replaced by steel. Also the period has seen the development of many innovations in the design of the structures of ships, some of which have survived, and some of which have been outmoded.

A study of Lloyd's Rules over the period under consideration shows that they reflect these changes. Lloyd's Register of Shipping have not resisted new ideas in the field of structures, but it is well to bear in mind that they have a great responsibility to the shipping industry, and, in consequence, they are obliged to move cautiously in order to ensure that the safety of ships is not jeopardized.

It is interesting to note that the early Rules were simple in nature, there being only one table devoted to the scantlings of the structure. It is clear that, as knowledge of the behavior of ships' structures developed, the Rules should become more extensive, in order to treat different sizes and types of ships fairly from a structural-strength point of view.

Mr. Murray traces in his paper the various changes in the Rules, and the relation between these changes and experience and research. We might make two comments here; firstly, with regard to the revision of the Rules in 1948, when the basis for the determination of scantlings was almost completely revised. Mr. Murray states that this revision brought about an increase of about 10 per cent in the section modulus of ships, and this was done because the development of certain features, such as welding, had led to higher bending moments. The other point about which comment might be made concerns the introduction in the Rules of measures to take into account departures from average design arrangements. The theory underlying that alteration is explained fully in the appendix to the paper, and reference is also made to the correction for speed, which has been introduced into the Rules, in vessels whose speed-length ratio exceeds 0.70.

It is evident from Mr. Murray's paper that Lloyd's Register of Shipping are still continuing the difficult task of producing more economical and efficient ships' structures; and to those who are not privileged to see the inside of these matters, it should be instructive in giving insight into the manner in which this great organization is endeavoring to bring about the co-ordination of research and experience for the benefit of those who earn their living by the sea.

One further note concerning Lloyd's Rules is that, as was stated in the issue of this journal for August, 1955, Rules have now been adopted for ships intended to carry ore, *i.e.*, for that type of ship in which the machinery is aft, and there are two longitudinal bulkheads forming central ore holds, with side tanks for either water ballast or oil.

We had occasion to refer, last year, to some interesting experiments on a ship-model among waves, which were carried out in America. Now, we have to record a series of experiments, of a similar type, which were carried out in the Mejiro Experiment Tank, and which have been published in a paper entitled "Model Experiments on the Strength of Ships Moving in Waves," read by Mr. Y. Akita and Mr. K. Ochi before the American Society of Naval Architects and Marine Engineers in November last.

These experiments differ from the American experiments in that the model was constructed of brass, being 6 meters long and having a simplified form, which consisted of wall-sided sections and a flat bottom. The model was towed in the tank at various speeds among waves of different dimensions, and was free to pitch and heave. Bottom pressure gauges were fitted, and the stress in the deck at amidships was measured by means of an electrical resistance strain gauge. The experiments indicated that the bottom pressures increased with more height, and that there was a slamming zone in a well-defined speed range. The peak pressures occurred at about $V/\sqrt{L}=1.2$, and, corresponding to these high pressures, there were high values of the deck stress. Defining the impact stress ratio as the ratio of the impact stress to the sagging stress at zero ship speed, the experiments showed that this ratio could be very high, reaching a value of 4 in the case of an L/26 wave at a ship speed of 1.4 meters per second. This means that, initiatively, the dynamic stress was four times the static stress.

This high figure in the experiments was largely due to the excessive flatness of the bottom of the model. Nevertheless, it gives some indication of the magnitude of the stresses which can be achieved at positions in the structure remote from the point of application of the slamming blow. One further interesting result from these experiments was that the stress, excluding impact stress, increased with ship speed indicating that the disturbance created by the ship had an influence on the profile of the waves in which the model was moving.

The development of buckles in the plating of ships subjected to compression is a matter which has achieved considerable importance in recent years. An approach to this problem has been made by Mr. M. R. Horne, in a paper included in the Transactions of the Institution of Naval Architects. This paper, entitled "The Progressive Buckling of Plates Subjected to Cycles of Longitudinal Strain," described a theoretical approach to the problem when plates are subjected to alternate compressive and tensile strains. It is shown that when plates, with or without initial de-

formations, are contained between longitudinal members which remain elastic, the development of large buckles should be prevented. The fact that plates in such circumstances can show permanent deformation is explained by the fact that the plate suffers a small permanent deformation after each application of load, whether compressive or tensile. In this way, large deflections build up. The complete theoretical solution of this behavior is difficult, and, for convenience, charts were prepared by the author which show the conditions under which large deflections can arise and the maximum range of stress in the supporting members if they are to be avoided.

It is also shown that when several panels are placed in series in the longitudinal direction, without substantial cross-members, much lower ranges of stress may cause one of the panels to develop large deflections, the other panels remaining unaffected.

The use of the corrugated bulkhead is now common in tanker practice, and may even eventually find its way into regular use in other types of ships. It is of interest, therefore, to draw attention to the results of a very complete investigation into this subject, which have been made known during the past year. This work is to be found in a paper by Mr. J. B. Caldwell, B.Eng., Ph.D., read before the Institution of Naval Architects in June last, and entitled "The Strength of Corrugated Plating for Ships' Bulkheads." In this work, the strength of corrugated plate used as beams was first investigated. For this purpose, both steel and aluminum alloy were used, and provided that failure did not take place by local instability, it was shown that the classical theories of elastic and inelastic bending described the behavior of the sheets adequately.

Further tests showed that the shape of the trough has an important effect on the stability of the sheet. Tests to destruction were made on eight model bulkheads under water pressure. The results showed that the non-buckling behavior of ships' bulkheads can be predicted by a simplified theory. The failing strength may, however, be seriously impaired if due precautions are not taken to avoid instability.

Some points in connection with the shape of trough might be quoted. It was shown that the pitch of the corrugations should be as large as possible, and that a deep trough in thin plating is more efficient than a shallow trough of equal section modulus in thicker plating. A rectangular trough is more efficient than a trapezoidal or triangular profile of equal section modulus.

The investigation showed quite clearly that, from a weight point of view, the corrugated bulkhead is much lighter than the traditional arrangement with plating and stiffeners.

Research has continued in the field of the application of aluminum alloys to ships, and we refer to work along this particular line which has been published in two papers during the year.

The first of these was by Mr. W. Muckle, M.Sc., Ph.D., entitled "The Influence of Proportions on the Behavior of Partial Superstructures Constructed of Aluminum Alloy," and presented to the Institution of Naval Architects in June last. The paper gave the results of an investigation on superstructures carried out for the Aluminum Development Association. This research consisted of model experiments on composite structures, consisting of a steel main structure with aluminum structures of varying lengths, breadths and thicknesses added. The superstructures were of a simple nature, without stiffening, but sufficiently rigid to prevent transverse distortion, and were tested with a centrally applied load, strains being measured by electrical resistance strain gauges. The experiment showed that, unless a superstructure was some 50 per cent or more of the length of the complete structure, the stress distribution at the midlength was not according to bending theory, and the superstructure was less than 100 per cent effective. The investigation included the development of an approximate theory to explain the experimental results.

The second of these papers—"An Analysis of the Behavior of Riveted Joints in Aluminum-alloy Ships' Plating"—was read before the North-East Coast Institution of Engineers and Shipbuilders, in November last, by Mr. A. R. Flint, B.Sc., Ph.D., and gave the results of an investigation on riveted joints in aluminum-alloy plates. This work was undertaken for the British Shipbuilding Research Association. The tests carried out included experiments on cold-driven N-6 rivets and hot-driven mild-steel rivets. The specimens were designed to indicate the influence of driving method, edge distance, rivet pitch, row spacing and rivet-diameter to plate-thickness ratio. The tests included both lap and cover-plate types of joints and the question of caulking of joints and watertightness were considered. The actual results were compared with theoretical predictions.

The paper gives some guidance on the design of joints in aluminum plating on the basis of the experimental results, and a very useful table is included, giving the particulars for optimum joints in N-6 plating for a wide range of thickness and for a number of rows of rivets from one to six. Not the least interesting part of this table is that dealing with joint efficiency, where it is shown that little improvement in efficiency is to be expected by increasing the number of rows beyond four. This is particularly true for thinner plating. This piece of work will be exceedingly useful to designers of aluminum-alloy structures, as it has filled a gap in knowledge on this subject and has cleared up some misconceptions, particularly with regard to the spacing required for watertight work.

Turning now to the question of welding, which still continues to be a subject of research. Before dealing with the publications in the more specialized aspects of this subject, we would draw attention to a paper of general interest to shipbuilders, entitled "The Economic Proportion of Welding in Shipbuilding," read by Mr. W. R. Mellanby before the North-Eastern Branch of the Institute of Welding, in April 1955. In it was contained a detailed comparison of the costs of construction of similar 10,000 to 12,000-ton deadweight ships, riveted and welded. The type of ship

considered was the two-deck type, for the carriage of dry cargo.

The investigation has shown that the welded ship is generally more expensive than the riveted ship, but, on the other hand, there is a saving in weight, which enables the cargo-carrying capacity to be increased. The conclusion arrived at is that the most economic design for a cargo vessel of this type is one with all-welded double bottom and both decks, and with welded shell butts and seams. If, however, large sections cannot be prepared at ground level, riveting of the shell is to be preferred.

We would also direct attention to a most interesting and useful theoretical study of stress distribution in fillet welds developed by Mr. P. J. Palmer, M.Sc., Ph.D., D.I.C., of the University of Birmingham, and which has been described in a paper entitled "Stress Distribution in Side Fillet Welded Plates," published in the *British Welding Journal* for February, 1955. The theory deals with double cover strap butt-jointed plates. The method adopted was to assume a general stress distribution such that the equilibrium conditions and the boundary conditions are satisfied. The unknown coefficients are determined by applying the principle of minimum potential energy. The results of the theory were compared with experimental results, and the agreement was good. Both the theory and the practical results show that considerable variations in longitudinal stress occur across the main and cover plates of joints of this type.

Interest continues to be shown in the welding of aluminum alloys, and we would refer to three papers published in the *British Welding Journal* for October, 1955. These contributions are as follows:—"Economics of the Joining of Aluminum and its Alloys by the Inert-gas Shielded-arc Welding Processes," by Mr. J. G. Young, B.Sc., "A Simple Cracking Test for Use with Argon-arc Welding," by Mr. P. T. Houldcroft, B.Sc., and "Heat Flow in Argon-arc Welding," by Mr. R. L. Apps, B.Sc., and Mr. D. R. Milner, B.Sc., M.Sc. Some of the problems referred to in these papers apply to metals other than aluminum, and to structures other than those of ships. Nevertheless, there is a good deal to be found in these contributions which will be useful to the shipbuilder contemplating the use of aluminum.

On the subject of aluminum and its alloys, it is of interest to note that the centenary of the discovery of aluminum was celebrated during 1955, and, to commemorate the event, the Aluminum Development Association organized an exhibition in the Royal Festival Hall, London, from the 1st to 10th June.

The June Number of *Welding and Metal Fabrication* contained much useful information on aluminum particularly with reference to its use in shipbuilding. In this issue is to be found articles entitled "The Present Position of Aluminum in Shipbuilding," written by Mr. E. G. West, Ph.D., B.Sc., F.I.M.; "A Review of Aluminum Alloys as Engineering Materials," by Mr. J. F. Handforth, M.Sc.; an account of aluminum in Norwegian shipbuilding, where extensive use is made of the metal arc-welding process; a description of the fabricating of the aluminum superstructure of the *Sunrip;* and, finally, an article entitled "Sigma Welding in Ship Construction—Automatic Assembly of Light-alloy Structures," by Mr. D. B. Tait, B.Sc. (Met.), B.Sc. (Eng.).

To conclude this section dealing with structural strength and welding, we mention the symposium on "The Use and Welding of Aluminum in Shipbuilding," which was held in London, in the lecture hall of the Institution of Civil Engineers, on the 7th and 8th December, 1955. Altogether, more than 20 papers were read on various aspects of welded aluminum in shipbuilding. It is impossible to give any detailed information of this comprehensive series of papers, but it is sufficient to say that they were grouped into four sections, *viz.*:—(1) Welding processes and technics, (2) materials and design (3) applications and (4) applications and economic factors.

The whole symposium forms an invaluable set of papers, which will, no doubt, be studied with great interest in the future.

RESISTANCE AND PROPULSION

Much useful information has come to light during the past year in this very important branch of naval architecture, and we shall refer to the outstanding contributions which are to be found in technical literature.

We have had occasion to refer, in the past, to the very important series of experiments which were carried out on the *Lucy Ashton,* and we have discussed the results as they have been made available by the British Shipbuilding Research Association. The fourth and final paper in this series—"B.S.R.A. Resistance Experiments on the *Lucy Ashton,* Part IV.—Miscellaneous Investigations and General Appraisal"—was written by Mr. S. Livingston Smith, C.B.E., D.Sc. (Eng.), and was presented at the last spring meetings of the Institution of Naval Architects. This paper was concerned with certain miscellaneous investigations which were made during the course of the experiments, and included the effects of fouling on resistance corresponding to various periods of immersion at different times of the year, acceleration and retardation trials, from which the virtual mass of the ship was determined, and Pitot transverses in the ship's frictional belt, giving velocity distributions for various hull-surface conditions.

Some indication of the increase in the resistance due to fouling is of interest. A light covering of barnacles, about ⅛ in. high, on the flat of the bottom, in association with a band of fine grass on the sides, gave rise to increases in resistance of 42 per cent, 29 per cent and 21 per cent, at speeds of 6, 12 and 14 knots, respectively. These increases in resistance were due to a lying-up period of 58 days at a buoy in the Gareloch. The whole series of results now available for the *Lucy Ashton* provides a comprehensive set of full-scale data, which should throw considerable light on many of the problems of ship resistance.

Two researches on the frictional resistance of ships have been recorded during the year. The first of these,

entitled "Frictional Drag of Smooth and Rough Ship Forms," by Ir. A. J. W. Lap, was presented at the autumn meeting of the Institution of Naval Architects, held in Yugoslavia during September, 1955. The author of this paper has examined very thoroughly all the existing formulae for pipes and flat plates, and has come to the conclusion that the frictional-resistance coefficients of pipes, plates and ship forms can be represented by the general formula.

$$\frac{\kappa\sqrt{2}}{\sqrt{\zeta}} = \log\frac{Re\sqrt{\zeta}}{A_n} + C$$

where ξ is the total frictional-resistance coefficient and Re is Reynolds number. A_n is a coefficient whose values differ for pipes and plates, as does the value of the coefficient κ, which is also affected by roughness.

The second paper, entitled "Investigation of the Resistance of an 18-ft. Plank," by Mr. J. F. Allan, D.Sc., and Mr. R. S. Cutland, was published in Volume 99, Part I, of the Transactions of the Institution of Engineers and Shipbuilders in Scotland. The investigation covers experiments on a plank with a smooth surface, a rough surface, and a surface arranged to represent a ship's hull with plate edges, rivets, welds, etc., the work having been carried out for the British Shipbuilding Research Association. The plank was 18 ft. in length, with a 12-ft. parallel portion amidships, 3 in. in beam and 29 in. in depth, and was constructed of aluminum alloy. For the smooth plank, the surface was polished with a cutting paste; while for the rough plank, emery powder of 0.01 in. grain size was used. The drag of the plank was measured by the momentum-drag Pilot traverse method. The paper gives very complete results of these tests, which add much useful knowledge on a very difficult and elusive subject.

Brief reference should be made to the results of full-scale trials on two ships. The first of these ships was a motor-driven cargo liner of 9,500 tons deadweight, and the results are given by Professor G. Aertssen in his paper entitled "Sea Trials on a 9,500-ton Deadweight Motor Cargo Liner," which was read as a joint contribution to the Institution of Naval Architects and Institute of Marine Engineers, in April, 1955.

The second series of tests appeared in the form of a paper entitled "Measurements on M.V. *Rijeka*, with their Attempted Practical Application," which was presented by Professor Ing S. Silovic and Ing M. Fancev at the last autumn meeting of the Institution of Naval Architects, held in Yugoslavia.

No attempt will be made here to discuss the results of either of these sets of tests in detail. It is sufficient to say that the naval-architecture profession is indebted to the authors, and to the organizations which they represent, for making available these very valuable data, which should assist in bringing nearer the accurate correlation of model results and the performance of full scale ships.

On the question of marine propellers, there have been some valuable contributions during the year. First, we would refer to a paper on "The Optimum Diameter of Marine Propellers—a New Design Approach," by Professor L. C. Burrill, M.Sc., Ph.D., of King's College, Newcastle-on-Tyne, which was read before the North-East Coast Institution of Engineers and Shipbuilders in November last.

In this investigation, the author first of all checks existing theory for determining the thrust, torque and efficiency, against standard series results on model propellers, and finds good agreement. The optimum diameter and optimum pitch distribution were then examined for propellers in a single-screw variable wake stream.

It was found that, when due account of the difference in wake velocity, associated with smaller diameters which cut off a portion only of the basic pattern, there is advantage to be gained by reducing the diameter materially, as compared with the open-water optimum condition. For example, results in the paper indicate that propellers could certainly be as small as 0.95 of the optimum diameter, and probably even less, and yet not suffer in efficiency. The conclusion is an important one, in view of the increase in diameters of propellers for large tankers at the present day.

Professor Burrill gives a design procedure which very clearly sets out this new approach to the design of propellers without the use of a standard series chart. The important assumption upon which the whole investigation rests is that the blade sections can be given the center-line camber appropriate to the minimum profile drag for the lift coefficient (C_L) value under consideration.

It is worthy of note that propellers designed on this new basis have different distributions of thrust from conventional designs, *i.e.*, the inner sections have higher thrusts and the outer sections less thrusts than normally. This has the effect of reducing the bending moments on the propeller blades. It is quite clear from this paper that new ground has been opened up, and propeller designers would do well to consider this new approach to the problem.

Propeller design has also been the subject of a paper by Lieut.-Commander M. K. Eckhardt, U.S.N., and Mr. W. B. Morgan, entitled "A Propeller Design Method" and read before the American Society of Naval Architects and Marine Engineers, in New York, during November last. The design method is based on hydrodynamic principles and is developed around the circulation or lifting-line theory. It includes a pitch correction, which complements lifting-line theory with lifting-surface theory. Two examples are given in this paper, one concerning a non-optimum free-running propeller and the other a wake-adapted propeller. A comparison with Lerbs' induction-factor method shows that the method is accurate enough for the type of propeller being considered.

The two examples given were widely different. The first was a propeller for a 35-knot twin-screw vessel, each screw absorbing 33,800 S.H.P. at 300 r.p.m. and producing a thrust of 210,500 lb. The second example was that of a 21-knot single-screw merchant vessel,

which absorbs 17,500 S.H.P. at 102 r.p.m. and produces a thrust of 237,000 lb. For the latter case, a wake-adapted propeller was designed to meet the conditions occurring behind the stern of the ship. The authors claim that this method of design reduces the amount of labor required by methods such as Lerbs' without affecting the accuracy. The method has not only proved to be successful as compared with other design methods, but also by comparison with numerous model tests.

The type of blade section employed in propellers has a considerable influence on their performance. It is of interest, therefore, to record some results from the National Physical Laboratory on this subject. Those who wish to study this work in detail will find it recorded in a paper, read before the North-East Coast Institution of Engineers and Shipbuilders in January, 1955, and entitled "Some Effects of Blade-section Shape on Model Screw Performance," and the authors of which were Mr. A. Silverleaf, B.Sc., and Mr. T. P. O'Brien.

Five screws, having different blade-section shapes, were designed for a specified loading and compared with a standard screw. These six model screws were tested in a water tunnel under cavitating and non-cavitating conditions. Under non-cavitating conditions no significant difference in performance was found, but under cavitating conditions, causing thrust breakdown, differences in performance were measured for the different types of blade section.

Some of the conclusions from this investigation might be quoted. Firstly, it is considered that the design method given by Hill is a reliable one, but that the lift-camber relation quoted by Hill for a parabolic mean-line requires further investigation. Under non-cavitating conditions, provided that the blade sections have good lift and drag characteristics, variations in section shape have little effect on efficiency. Under cavitating conditions, changes in blade-section shape can cause significant changes. Comparison of thrust-breakdown conditions, and thrust and efficiency reduction factors, suggests that shockless entry designs are inferior to those with a small amount of positive incidence.

A contribution to the singing-propeller problem has been made by Mr. S. G. Lankester, B.Sc., A.R.C.S., and Mr. W. D. Wallace, B.A., R.C.N.C., in a paper read before the North-East Coast Institution of Engineers and Shipbuilders in April, 1955. In this contribution, entitled "Some Investigations into Singing Propellers," there were reviewed some cases of singing propellers investigated by the Admiralty in the past few years.

The authors define singing, so far as propellers are concerned, as being that condition where one or more distinct frequencies are audible in the general noise spectrum emitted by the propeller. It is considered that there are four possible causes of singing, and these are eddy-shedding from trailing edge or tip, fluctuations in flow conditions at the leading edge, a phenomenon similar to aircraft-wing flutter, and self-excited vibrations caused by bearing friction.

The modifications made to a number of propellers, in an attempt to cure singing, are listed in the paper, together with the results obtained. It is recommended that further research should be carried out on this problem by experimenting on singing vanes, by the use of hot-wire anemo-trailing edges of such vanes, in order to locate and measure the exciting forces, and by full-scale trials, in which further modifications are made to singing propellers. It is also considered that an extension of the theory of flutter is needed, while further experiments on singing model propellers would be useful.

It is more than 25 years since the Voith-Schneider propeller was introduced, and, since then, there has not been a great deal of information published on this method of propulsion. Naval architects who have to deal with special propulsion problems, and may be thinking in terms of this particular type of propulsion, will welcome a paper read by Dr. Hans F. Mueller before the American Society of Naval Architects and Marine Engineers, in May, 1955. This paper, entitled "Recent Developments in the Design and Application of the Vertical-axis Propeller," discusses the history of this method of propulsion, and also the hydrodynamical principles on which it operates. The method of plotting comparative efficiencies which relates the propeller efficiencies to the ideal jet efficiency, is used, and the term "degree of perfection" is introduced. Detailed information is given of the latest design of two 1,000-H.P. propellers for a United States Army towboat. The results of stress and deflection tests are given in the paper.

Another form of propulsion, which is now rather uncommon, is the paddle wheel. Recently, however, details of model experiments on paddle wheels were published in a paper entitled "Paddle Wheels—Part I, Preliminary Model Experiments," which was read by Mr. H. Volpich, B.Sc., and Mr. I. C. Bridge, B.Sc., before the Institution of Engineers and Shipbuilders in Scotland, in February last.

The experiments described in this paper were carried out in the tank of Messrs. William Denny & Brothers, Ltd., Dumbarton. Experiments of this nature necessitated the design of special apparatus, as normal tank equipment is usually limited to the testing of screw propellers only. Two wheels were tested, the larger being 3.4 ft. in diameter and having floats 2.5 ft. long, while the other wheel was 1.7 ft. in diameter and had floats 1.5 ft. long. The number of floats was nine in each case. Both radial wheels and feathering wheels were tested.

It is of interest to note from these experiments that the efficiency of paddle wheels can be quite high—as much as 80 per cent in some cases. The conclusion reached from the preliminary tests is that the problem of paddle propulsion is much more complicated than might at first be expected. The revolutions per minute, it would seem, have an effect which makes slip an incomplete criterion of the performance, and rules out any possibility of a simple theoretical approach on a momentum basis. It is proposed to extend the investigation on feathering wheels to immersions

of up to 30 per cent with six, nine and eleven floats. The results of these further tests will be awaited with interest.

A paper of considerable interest, entitled "Ship Speeds in Irregular Seas," was read by Mr. Edward V. Lewis before the American Society of Naval Architects and Marine Engineers, in November last. This work is based, in part, on the research in sea-keeping qualities of ships, carried out at the Experimental Towing Tank, Stevens Institute of Technology, and sponsored by the Bureau of Ships. The effect of ship proportions on the speeds at which severe pitching and heaving motions will occur is demonstrated. Experimental confirmation is given of the method of representing ship motions in irregular seas, by means of the linear superposition of responses to the component waves in a seaway. The most objectional ship motions—involving shipping of water, slamming and high accelerations—result from responses to the wave components which approach synchronism with the ship's natural periods of oscillation.

In concluding this section on resistance and propulsion, mention should be made of the work proceeding under the auspices of the British Shipbuilding Research Association. The methodical series of model experiments on ocean-going vessels continues. This program includes the study of the effect of changes in the longitudinal position of the center of buoyancy, bilge radius and draft. Tests have been carried out on forms having block coefficients varying in steps from 0.65 to 0.80.

During the past year, a report was issued to member firms presenting the results of propulsion tests on a model of a redesigned 0.75 block-coefficient series. The object in the tests was to compare the propulsive performance of the redesigned model with that of the original.

SHIP MOTION AND VIBRATION

These two aspects of naval architecture have had some attention devoted to them during the past 12 months. First, we shall consider the stabilization of the rolling motion of ships. This is a problem to which many inventors have devoted their energies in the past, and the Transactions of the Institution of Naval Architects contain the records of attempts to produce ships which do not roll. Mention may be made of such devices as the gyroscope and the anti-rolling tank.

Of all the inventions in the field of stabilization, the only one which appears to have survived, and is, in any sense, in general use today, is the activated fin. It is of interest, therefore, to focus attention on a paper dealing with one of the most successful activated fins—the Denny-Brown stabilizer. This paper, entitled "Experiences in the Stabilization of Ships," was read by Sir William Wallace, C.B.E., F.R.S.E., before the Institution of Engineers and Shipbuilders in Scotland, in January, 1955.

We shall not attempt to go into any great detail concerning Sir William's paper; it is sufficient to say that it gives a history of the device up to the present, and explains the difficulties which were encountered and how they were overcome. Details are given of some typical fin installations, including the most up-to-date flap-type fin. Indicative of the efficiency of the fin is the rolling record for the *Chusan*, reproduced in the paper. In a journey from Hong Kong to Singapore, the angle of roll without the fin in operation was from 9 to 10 deg., out to out. With the fins in operation, this was, for all practical purposes, reduced to zero.

Stabilization has also been the subject of investigation in the United States of America, and we would mention, in this connection, the paper entitled "The Stabilization of Roll," read by Mr. J. H. Chadwick, Jr., before the American Society of Naval Architects and Marine Engineers, in November last.

Mr. Chadwick considers the history of stabilization and deals with the theory in the light of modern knowledge. It is interesting to note that, while the activated fin is taken to be the most suitable means of stabilization at speeds above about 15 knots, it is believed that tanks and gyroscopes should not be dismissed for low-speed applications. The technical possibility of stabilization is an established fact, and, in future, greater emphasis will be placed on the economic aspects of the problem.

It is considered by Mr. Chadwick that the methods developed by St. Denis and Preston (to which we have made reference in previous years) will play an important part in future investigations of roll stabilization. Several modes of control are possible, but, at least, some feed-back control should always be used. Arguments for feed-ahead control and predictions have been negated by advances in control technology.

Both these papers, which deal virtually with the same problem, are worthy of careful study by specialists in this particular field.

An interesting investigation has been made during the past year by Mr. A. J. Sims, O.B.E., R.C.N.C., and Mr. A. J. Williams, R.C.N.C., and particulars of which appeared in a paper entitled "The Pitching and Heaving of Ships," which was read at the autumn meeting of the Institution of Naval Architects in Yugoslavia. Pitching and heaving are most important problems in naval vessels, especially in aircraft- carriers. In this work, the theories of pitching and heaving have been re-examined. The static amplitudes of pitch and heave were first calculated. It is shown that, so far as pitching is concerned, the static amplitude is dependent upon the maximum surface wave-slope, the ship-length/wave-length ratio, and a factor which depends upon time and the shape of the water-plane of the ship.

The influence of the timing factor on the static amplitude is considered, and this, of course, depends upon the undamped free oscillation period of the ship in relation to the period of the exciting force. An equation is deduced, giving the dynamic amplitude in terms of the static amplitude, and curves of that ratio are plotted for different values of the tuning factor and the damping coefficient. The theory is compared with model experiments on an aircraft-carrier model in waves having a height of L/40.

The theoretical and experimental curves were adjusted to correspond as nearly as possible, and, in this way, a damping coefficient was deduced. Heaving is treated in the same way, and the combined effects of heaving and pitching are treated. One of the important conclusions from this investigation is that wave slope and height are unsuitable parameters for analysis of pitching and heaving data by the standard oscillation theory, and that these parameters should be qualified by conversion factors which depend upon the geometry of the ship's form and the ratio of ship length to wave length.

Professor B. V. Korvin-Kroukovsky, in the United States, has also dealt with the problem of the pitching and heaving of ships, in a paper read before the American Society of Naval Architects and Marine Engineers, in November last, and entitled "Investigation of Ship Motions in Regular Waves." In this treatment of the problem, the differential equations of motion were first formulated, by equating the sum of the intertial and hydrostatic restoring forces to the hydrodynamic forces resulting from the velocities and accelerations of the water. The solution for the hydro-dynamic force is then carried out for a heaving and pitching ship, advancing with or against a train of regular waves. The solution is compared with the results of model experiments, and it is shown that the apparent damping is exaggerated. It is considered that the method described in the paper could be used to modify lines in the early stages of a design, and so obtain a more seakindly ship. These lines could then be tested in the towing tank for resistance.

Specialists in the field of ship vibration will welcome the very complete investigation carried out in America on the *Gopher Mariner*. The results of this investigation are to be found in a paper entitled "Hull Vibration Investigation of S.S. *Gopher Mariner*," by Mr. Raymond T. McGoldrick and Mr. Vito L. Russo, which was read before the American Society of Naval Architects and Marine Engineers last November. The tests described here were a joint undertaking of two of the Society's Research Panels, *viz.*, Panel S-6 of the Hull Structure Committee and Panel H-8 of the Hydrodynamics Committee.

GENERAL TOPICS

In the concluding section of this article, brief reference will be made to some matters of general interest. First of all, with regard to shipbuilding and ship-repairing, we would mention two papers read at the autumn meeting of the Institution of Naval Architects, in Yugoslavia, *viz.*, "Organizational Developments of the Shipyard Uljanik," by Ing. Djuro Stipetic, and "The Recent Development of the Shipyard at Split," by Mr. M. Stanger and Mr. S. Ercegovic.

The former of these yards, *viz.*, Shipyard Uljanik, at Pula, Yugoslavia, is over 100 years old, and has gone through many changes in its time. At present, there are employed there about 2,500 people. The shipyard at Split was moved to its present site in 1925. The works passed through several hands, and eventually, in 1948, extensions to the yard were tackled. The task may be stated as making the shipyard capable of working 30,000 tons of steel annually, and to be able to handle ship repairs up to 2,000 tons of steel per annum. It is of interest to note that the present yard covers a site of 100 acres, has five berths and employs 4,000 people.

On the ship-repairing side, we would refer to a paper read before the American Society of Naval Architects and Marine Engineers, in May, 1955, and entitled "A Method for Increasing the Efficiency of Ship-repair Yard Operations." This paper, which was written by Mr. Robert Schor, is a plea for the application of modern production planning principles to a repair shipyard, in which the work is non-repetitive and, to a large exent, unpredictable. This contribution should prove of interest to those engaged in this very important branch of the shipping business.

While on the subject of shipbuilding and repairing, we might mention the paper read by Mr. Ivor E. King, C.B.E., R.C.N.C., before the Institution of Naval Architects, in April, 1955, and entitled "Forty Years of Change at Portsmouth Dockyard, with some notes on Dockyard Organization." As indicated, this paper deals with the changes in H.M. Dockyard at Portsmouth over the past 40 years, and gives an account of dockyard organization generally.

Improvement in habitability in naval ships has been dealt with in a paper entitled "Habitability of Naval Ships," which Mr. D. S. Berres read before the American Society of Naval Architects and Marine Engineers in November last. All the problems involved in providing suitable accommodation in naval ships are discussed including such matters as compartment size and arrangement, traffic control, furniture design, air-conditioning, lighting, deck covering, color schemes and noise. Some recommended arrangements are given.

Attention has again been drawn to the question of deterioration of cargoes during transit, in a paper entitled "Deterioration of Motorcars during Shipment Overseas," which Mr. E. A. Shipley, B.Sc., read before the North-East Coast Institution of Engineers and Shipbuilders, in February, 1955. It may come as a surprise to many that this is a most serious problem, which affects both car manufacturers and shipowners alike. A great deal has been accomplished by manufacturers in attempting to avoid corrosion of components and steelwork, but, apparently, a most important problem still to be tackled adequately is the conditions in the holds of the ships in which the cars are carried. The author makes a plea for the greater use of mechanical ventilation, with air-drying equipment, although it would appear that mechanical ventilation only, with proper records of cargo-hold temperatures and dew-points, might be effective in preventing damage.

Naval life-saving equipment has been discussed in a paper, by Mr. W. J. Holt, R.C.N.C., entitled "Recent Developments in Naval Life-saving Equipment," read

before the Institution of Naval Architects, in April, 1955. This development has taken place largely since the Second World War, and is, to a considerable extent, due to experiences in the war. It is clear that this is a matter which merits very careful consideration from many points of view, and, no doubt, the equipment produced for this purpose will depend, to a considerable extent, on the hazards which might be expected in any future war from modern weapons.

We have had occasion, in former years, to refer to cathodic protection of ships' hulls, and we draw attention to this problem again, by referring to a paper on "The Cathodic Protection of Ships against Seawater Corrosion," read before the Institution of Naval Architects, in April, 1955, by Mr. L. T. Carter, B.Sc., R.C.N.C., and Mr. J. T. Crennell, M.A. This contribution adds to the now quite considerable literature on the subject, and is worthy of study by those faced with corrosion problems.

CONCLUSION

An attempt has been made, in the various sections of this article, to indicate the most important results of research in the different branches of naval architecture and of shipbuilding during the past year. It is obvious that many problems still remain to be solved; it is equally obvious that the ultimate solution of many of the problems will not be brought about by any one individual organization or country alone. Only by the united efforts of all those interested in the design, construction and operation of ships will better and safer ships eventually be achieved.

NORMAN H. JASPER

TEMPERATURE–INDUCED STRESSES IN BEAMS AND SHIPS

THE AUTHOR

received his Bachelor's degree in Mechanical Engineering from the College of the City of New York in 1941, the Master of Science degree from the University of Maryland in 1950 and the Doctorate in Engineering from the Catholic University of America in 1956. He worked in the Scientific and Test Group at the Puget Sound Naval Shipyard from 1941 until his transfer to the David Taylor Model Basin in 1946 where he has worked in the fields of vibration and ship structural problems. Earlier papers by the author appeared in this journal in 1950 and 1951 on the application of statistical methods to the study of the stresses and motions of ships and in 1953 on "Ship Vibration Problems." He is a member of the ASME, Sigma Xi, and the Society of Naval Architects where he has been active in the work of its research panels.

The material in this article is taken from a report of the David Taylor Model Basin. The Society made no remuneration to the author at his request.

INTRODUCTION

It is a well-known fact that changes in the temperature distribution within a structure, such as a ship, may induce stresses even in the absence of external loads or restraints on the structure. Very few quantitative data are available which would indicate the actual stress and temperature variations that might be expected throughout a ship in service.[1, 2, 3]

Ships have broken in half while moored in still water. Mr. Arthur Gatewood of the American Bureau of Shipping has, in a private communication, called the author's attention to the observation that many major hull fractures in still water took place while the temperature was changing.* This, in itself is a significant observation. Its significance is accentuated by hull-girder strains measured over a 19-month period on the T-2 tanker *Esso Asheville*. These tests, which were sponsored by the S-10 Panel of the Society of Naval Architects and Marine Engineers, indicated the presence of a slow stress variation with a period of about a day which may be ascribed to the change in temperature distribution that occurs daily. These temperature-induced stress variations reached the same order of magnitude as the stresses due to the cargo loading of the ship; Figure 1 is a sample of oscillogram** of temperature-induced stress variations. The appendix discusses the methods utilized to make these measurements. There is sufficient evidence to indicate that an estimate of the temperature-induced hull-girder stresses† is warranted. It would therefore be advantageous to have a reasonably simple method which could be used by the designer to estimate the magnitude and distribution of possible temperature-induced stress variations.

It is the purpose of this report to develop and present such a method which may be readily applied to ships constructed of any combination of materials and subjected to any type of temperature distribution in the transverse section of the girder. How-

Superior figures refer to references at the end of the article.

*Examples are the fractures of the *Schenectady*, *Belle Isle*, and *Ponaganset*, all of which were afloat in still water at the time of failure.

**It is not to be inferred that the sample oscillogram is typical of everyday stress variations. Rather it illustrates fairly severe temperature-induced stress.

†Throughout this report we will be concerned with **changes** in stress corresponding to given **changes** in the distribution of temperatures. It will therefore be understood that the reference is to changes in temperature and changes in stress even though the word "change" is not specifically stated.

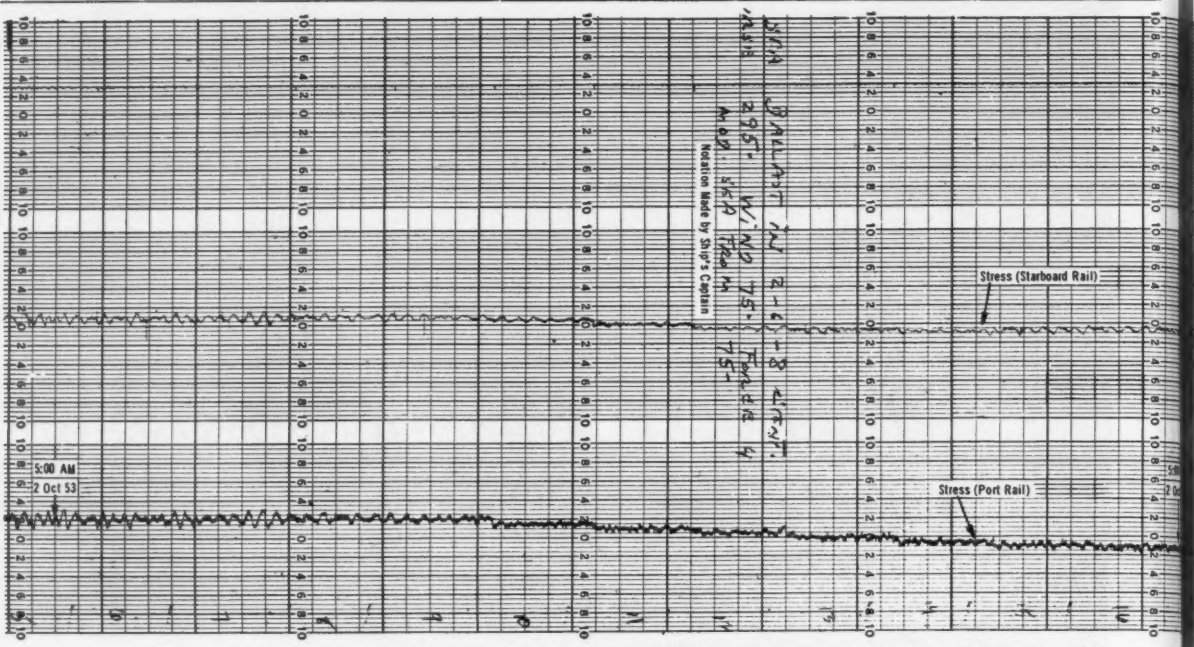


Figure 1. Sample Oscillograms Showing Diurnal Variations in the Mean Stress Level. (Incident to Temperature Changes)
The higher frequency stress variations are those incident to the passage of waves past the ship. The scale for stress is the same on all records.

ever, all transverse sections are assumed to have the same variation in temperature distribution.

Corlett [3] developed a method for computing temperature-induced stresses which, for symmetrical temperature distributions, and for box girders without transverse restraint such as are provided by transverse bulkheads, is practically identical with the method developed here. However, the author has been unable to follow the somewhat unusual reasoning employed by Corlett in his development, and furthermore, *Corlett's method is applicable only to cases where the temperature distribution is symmetrical with respect to the midplane of the ship and where there is no transverse restraint.* The basic theory* applied in this report is straight-forward and has been given by Timoshenko.[4]

The use of the theory is illustrated by application to the ship model tests conducted by Corlett, in order to check the validity of the method. Next the theory is applied to a T-2 tanker for both an assumed symmetrical and an unsymmetrical distribution of temperature changes.

STRESS DISTRIBUTION IN A BOX GIRDER DUE TO AN ARBITRARY TWO-DIMENSIONAL CHANGE OF TEMPERATURE GRADIENT

Let it be required to determine the stress distribution in a transverse section of an idealized ship girder** due to an arbitrary two-dimensional (in the plane of the transverse section) temperature gradient. The following assumptions will be made: (Refer to Figure 2)

1. The usual elementary theory of the bending of beams is assumed to be applicable. Thus in the initial development it is assumed that there are no restraints which would prevent deformation in the transverse planes and that $\sigma_y = \sigma_z = 0$. Later it will be shown that if such restraints were fully effective in a thin-walled box beam, for example, by providing closely spaced rigid diaphragms, all the stresses throughout the beam will be modified in the same ratio. The actual stress variations in a box beam should lie somewhere between these two extreme assumptions of full and zero transverse restraint.

2. The change in temperature distribution is identical in every transverse cross section of the beam.

3. There are no external moments or forces acting on the beam.

4. St. Venant's principle applies.

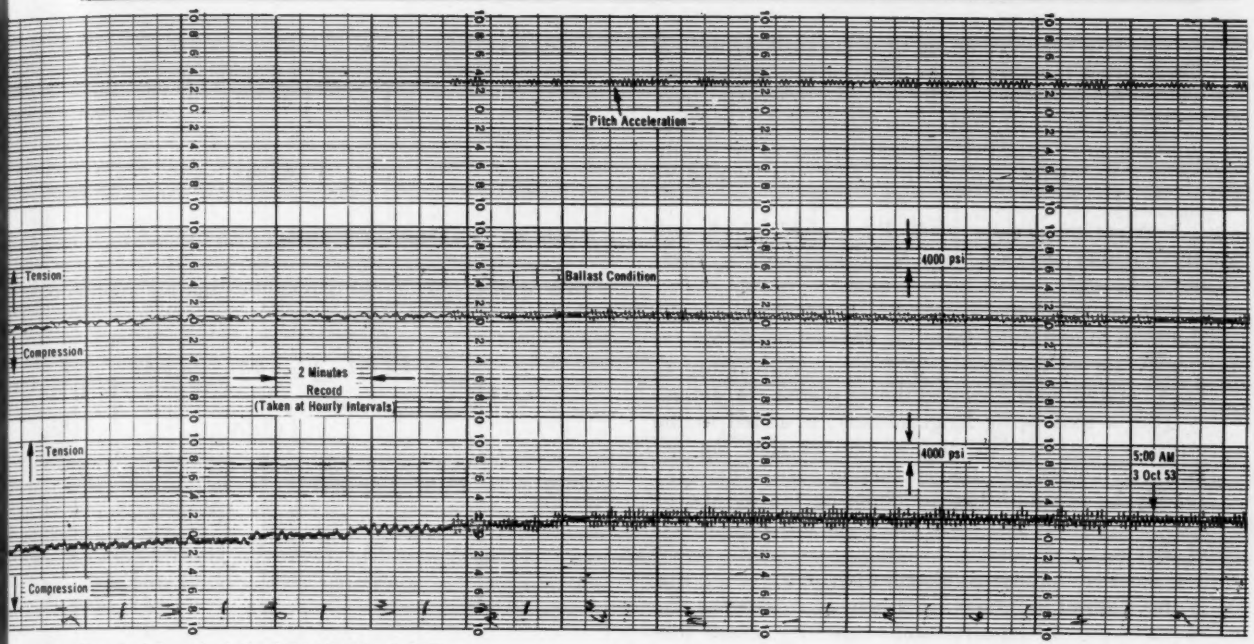
Figure 2 illustrates the beam and gives the assumed directions and the notation. The development here will follow that given by Timoshenko [4] for a thin plate. It follows from the assumptions that the stress distribution is the same at every transverse section sufficiently distant from the ends of the beam.

Equilibrium requires that the net moment, the total longitudinal force, and the total transverse shear force be zero in any direction at any transverse section. Since these requirements must be satisfied, the analysis of the problem may be conveniently carried out as follows for any homogeneous† beam. It is initially assumed that the girder

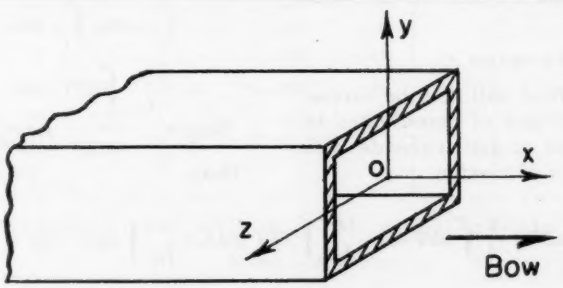
*Mar and Engel [5] used the same basic theory for solving the similar, but simpler, problem of temperature-induced stresses in a rectangular cantilever box beam subjected to a symmetrical distribution of temperature changes.

**The idealized girder is assumed to have a uniform cross section throughout its length and the connections between the sides of the box are assumed to act as though they were hinged.

†A nonhomogeneous beam may be treated as shown on pages 488-491.



Stress Change Due to Temperature Gradient is 7000 psi in 24 Hour Period (P)



x, y, z are the principal centroidal axes of inertia such that

$$\int Ey\,dA = \int Ez\,dA = \int Eyz\,dA = 0$$

x is the longitudinal axis
y is oriented in the vertical direction
α is the temperature coefficient of expansion of material
T is the temperature change, $T = f(y, z)$
A is the effective metal area of the cross section (homogeneous beam)
I_{oz} is the area moment of inertia about the axis oz
I_{oy} is the area moment of inertia about the axis oy
E is Young's modulus
e is the unit strain
σ is the normal stress in the x-direction
ν is Poisson's ratio
k is the ratio E/E_ε where E_ε is an arbitrary constant which is taken as E for steel in the illustrative examples

A_ε is the value of the integral $\int k dA$

$(I_{oz})_\varepsilon$ is the value of the integral $\int ky^2 dA$

$(I_{oy})_\varepsilon$ is the value of the integral $\int kz^2 dA$

σ_y, σ_z is the normal stress in the y and z directions, respectively

Figure 2. Nomenclature and Schematic Sketch of Beam.

The beam may be made up of different materials provided that the strain remains continuous.

acts as if it were made up of separate longitudinal fibers fitted together in the form of the ship girder. These elements are initially allowed to expand freely due to temperature changes. Stress fields are then imposed so as to satisfy the requirements of equilibrium, geometry, continuity, and the boundary conditions. Thus

1. A set of stresses σ_1 is determined which will prevent thermal expansion of all longitudinal fibers of the beam.*

2. A constant stress σ_2 is added which will make the total longitudinal force on the section zero.

3. A set of stresses σ_3 and σ_4, corresponding to pure bending in the xy and the xz planes respectively (the x axis being longitudinal), is added to the stresses σ_1 and σ_2 in such a manner that the net bending moment at the section is zero. This may readily be accomplished by choosing the oy and oz axes to coincide with the principal centroidal axes of inertia.

4. The resultant stress distribution $\sigma=\sigma_1+\sigma_2+\sigma_3+\sigma_4$ is the desired solution inasmuch as it satisfies the requirements of equilibrium, continuity, beam theory, and the boundary conditions, except near the ends of the beam. By St. Venant's principle the solution will be valid at sections distant from the ends of the beam, even though the boundary conditions are not completely satisfied at the ends of the beam.

HOMOGENEOUS BEAMS

The four steps just described will now be carried out analytically. Let the origin of coordinates be chosen such that the oy and oz axes coincide with the principal centroidal axes of inertia.

Step 1 $\quad \sigma_1=-\alpha ET$

Step 2 $\quad \sigma_2\int dA-\int \alpha ET\,dA=0$

Therefore

$$\sigma_2=\frac{1}{A}\int \alpha ET\,dA$$

Step 3 $\quad \int(\sigma_1+\sigma_2+\sigma_3+\sigma_4)\,y\,dA=0$

This expression is obtained by taking moments about the oz axis. Therefore

$$\sigma_3=y\left[\frac{-\sigma_2\int y\,dA-\int\sigma_1 y\,dA-\int\sigma_4 y\,dA}{\int y^2\,dA}\right]$$

since $\sigma_3=\frac{\sigma_{(y=c)}}{c}\,y$. Similarly

$$\sigma_4=z\left[\frac{-\sigma_2\int z\,dA-\int\sigma_1 z\,dA-\int\sigma_3 z\,dA}{\int z^2\,dA}\right]$$

since $\sigma_4=\frac{\sigma_{(z=c)}}{c}\,z$.

Since the oy and oz axes are chosen to coincide with the principal centroidal axes of inertia

$$\int z\,dA=\int y\,dA=\int\sigma_4 y\,dA=\int\sigma_3 z\,dA=0$$

$$\sigma_3=\frac{y}{I_{oz}}\int\alpha ET\,y\,dA \qquad \sigma_4=\frac{z}{I_{oy}}\int\alpha ET\,z\,dA$$

Step 4 $\quad \sigma=\sigma_1+\sigma^2+\sigma_3+\sigma_4$

Thus

$$\sigma=-\alpha ET+\frac{1}{A}\int\alpha ET\,dA+\frac{y}{I_{oz}}\int\alpha ET\,y\,dA+\frac{z}{I_{oy}}\int\alpha ET\,z\,dA \qquad [1]$$

This general expression for σ gives the stress distribution for any given change in the temperature distribution $T(y, z)$ provided that the assumptions stated at the beginning of this section may be considered applicable and that the beam is made of the same material throughout. If $T(y, z)$ and dA can be approximated by analytic expressions, then the integrals may be evaluated directly. For the more general and realistic cases (for example, unequal temperature gradients on the port and starboard sides of a ship) a numerical procedure will be found convenient; see Table 1. The schedule in Table 1 will enable one to determine the stress distribution in the transverse section.

The stress distribution in a T-2 tanker, under assumed conditions of temperature changes which are believed to be realistic, is shown in Figures 4 and 5. The numerical work for computing these stress distributions is given in Tables 2 and 3.

COMPOSITE BEAMS

For beams constructed of more than one material the preceding development must be modified** since both α and E may vary over the cross section. The same step-by-step procedure applied to the homogeneous beam is followed.

Step 1 $\quad \sigma_1=-\alpha ET$

Step 2 Now σ_2 must vary in proportion to E in order to make the strain the same in all fibers. Hence

$$\int\sigma_2\,dA-\int\alpha ET\,dA=0$$

and

$$\sigma_2=E\frac{\int\alpha ET\,dA}{\int E\,dA}$$

*This stress field will return all particles to the longitudinal position that they occupied before the application of the temperature changes.

**At first this development was carried out by the use of the concept of an equivalent homogeneous beam. Both the original and present approaches give the same stress distribution.

TABLE 1—*Computation of the Stress Distribution for the Steel-Aluminum Ship Model of Figure 3*

All units are given in inches and pound system.
Although more than two digits are carried through in the computations, no more than two significant figures are to be inferred.

	1	2	3	4	5	6	7	8	9	10	11	12	Theoretical 13	14	15	Experimental 16
	Item No	Area (ΔA) Actual	$\bar{y}$ of ΔA	$\bar{z}$ of ΔA	T at $\bar{y},\bar{z}$	$\alpha ET = -\sigma_1$ at $\bar{y},\bar{z}$	Col 2×6 $\alpha ET(\Delta A)$	Col 7×Col 3 $\alpha ET(\Delta A)\bar{y}$	Col 7×Col 4 $\alpha ET(\Delta A)\bar{z}$	From Cols 6, 2, 7 $(\sigma_1)+(\sigma_2)$ psi	From Cols 3, 8 (σ_3) psi	From Cols 4, 9 σ_4 psi	From Cols 10, 11, 12 $\sigma = \sigma_1+\sigma_2+\sigma_3+\sigma_4$	$\Sigma f = 0$ $\sigma(\Delta A)$	$\Sigma M = 0$ $\sigma(\Delta A)\bar{y}$	Experimental Stress Value
Aluminum $k = 1/3$	1	0.75	+13.08	–	51.5	+6597	4948	$+64.72\times10^3$	–	−5805	+1678	–	−4127	−3095	-40.5×10^3	−4323
	2	0.75	13.08	–	49.2	6174	4631	60.57	–	−5382	+1678	–	−3704	−2778	−36.3	−4122
	3	0.75	13.08	–	42.0	5391	4035	52.78	–	−4599	+1678	–	−2921	−2191	−28.7	−2957
	4	0.75	13.08	–	33.0	4227	3170	41.47	–	−3435	+1678	–	−1757	−1318	−17.2	−1770
	5	0.75	12.18	–	24.0	3074	2306	28.09	–	−2282	+1563	–	− 719	− 539	− 6.6	− 874
	6	0.75	10.18	–	20.0	2562	1922	19.56	–	−1770	+1306	–	− 464	− 348	− 3.5	− 314
	7	2.95	9.08	–	17.6	2254	6651	60.39	–	−1462	+1165	–	− 297	− 876	− 8.0	− 202
	8	0.61	8.68	–	16.8	2152	1313	11.40	–	−1360	+1113	–	− 247	− 151	− 1.3	− 179
	9	0.38	7.68	–	14.9	1909	725	5.56	–	−1117	+ 985	–	− 132	− 50	− 0.4	− 112
	10	0.69	6.18	–	12.0	1537	1061	6.55	–	− 745	+ 793	–	+ 48	+ 33	+ 0.2	+ 22
Steel $k = 1$	11	0.36	5.18	–	10.3	2086	751	3.89	–	+ 290	+1994	–	+2284	+ 822	+ 4.3	+2621
	12	2.92	4.86	–	4.8	972	2837	13.79	–	+1404	+1871	–	+3275	+9563	+46.5	+2576
	13	0.85	4.41	–	7.5	1519	1291	5.69	–	+ 857	+1697	–	+2554	+2171	+ 9.5	+2554
	14	0.38	3.41	–	5.5	1114	423	1.45	–	+1262	+1313	–	+2575	+ 979	+ 3.3	+2464
	15	0.75	+ 2.01	–	3.5	709	533	+ 1.06	–	+1667	+ 774	–	+2441	+1831	+ 3.7	+2173
	16	0.75	+ 0.01	–	2.0	405	301	0	–	+1971	+ 4	–	+1975	+1481	0	+1837
	17	0.75	− 1.99	–	1.2	243	183	− 0.36	–	+2133	− 766	–	+1367	+1025	− 2.0	+1366
	18	0.75	− 3.99	–	0.8	162	121	− 0.48	–	+2214	−1536	–	+ 678	+ 509	− 2.0	+ 851
	19	0.75	− 5.99	–	0.3	61	46	− 0.28	–	+2315	−2306	–	+ 9	+ 8	0	+ 336
	20	0.75	− 7.99	–	0.3	61	46	− 0.36	–	+2315	−3075	–	− 760	− 570	+ 4.6	− 269
	21	0.75	− 9.99	–	0.3	61	46	− 0.46	–	+2315	−3845	–	−1530	−1148	+11.5	− 986
	22	3.00	−10.89	–	0.3	61	183	− 1.99	–	+2315	−4192	–	−1877	−5631	+61.3	−1254
							37,523 $=\Sigma_7$	373,040 $=\Sigma_8$						−18,695 +18,422	− 146.5 + 144.9	

$E_{Steel} = 30\times10^6$, $(E\alpha)_{AL} = 128.1$, $(E\alpha)_{ST} = 202.5$, $\frac{E_s}{E_a} = 3$, $A_\epsilon = 15.79$, $(I_{oz})_\epsilon = 969.3$, $\sigma_2 = \frac{37523}{15.79}k = 2376k$ psi, $\sigma_3 = \frac{ky\ 373040}{969.3} = 384.9\bar{y}k$, $\sigma_4 = 0$,
N. Axis is 10.99″ above B.L.

Step. 3 In pure bending $\int \sigma_x dA = 0$. Also we locate the axes so that bending in yx and zx axes is uncoupled, and thus the strain is proportional to y or z respectively. Then

$$\int E\,y\,dA = \int E\,z\,dA = \int E\,y\,z\,dA = 0$$

This requirement may be thought of as the determination of the principal centroidal axis of an area comprised of area elements (EdA) which have the coordinates y, z of dA. Also

$$\sigma_3 = k_1 E\,y,\ \sigma_4 = k_2 E\,z$$

where k_1 and k_2 are constants.

$$\int (\sigma_1+\sigma_2+\sigma_3+\sigma_4)\,y\,dA = 0$$

Therefore

$$-\int \alpha ET\,y\,dA + k_1 \int E\,y^2\,dA = 0,$$

$$-\int \alpha ET\,z\,dA + k_2 \int E\,z^2\,dA = 0$$

$$\sigma_3 = \frac{E\,y}{\int E\,y^2\,dA}\int \alpha ET\,y\,dA,$$

$$\sigma_4 = \frac{E\,z}{\int E\,z^2\,dA}\int \alpha ET\,z\,dA$$

Step 4 $\sigma = \sigma_1+\sigma_2+\sigma_3+\sigma_4$

$$\sigma = -\alpha ET + E\frac{\int \alpha ET\,dA}{\int E\,dA} + \frac{E\,y}{\int E\,y^2\,dA}\int \alpha ET\,y\,dA + \frac{E\,z}{\int E\,z^2\,dA}\int \alpha ET\,z\,dA \qquad [2]$$

TABLE 2—*Computation of Stress Distribution in a T-2 Tanker*

Symmetrical distribution of temperature change. All units are given in inches and pound system.

Although more than two digits are carried through in the computations, no more than two significant figures are to be inferred.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Item No	ΔA	$\bar{y}$	$\bar{z}$	T deg F	αET	$\alpha ET \times \Delta A$	$\alpha ET(\Delta A)\bar{y}$	$\alpha ET(\Delta A)\bar{z}$	$\sigma_1+\sigma_2$	σ_3	σ_4	σ	$\sigma(\Delta A)$	$\sigma(\Delta A)\bar{y}$	$\sigma(\Delta A)\bar{z}$
1	89.0	240		70	13230	1.177×10^6	2825×10^5		−8116	6816		−1300			
2	178	238		70	13230	2.355	5605		−8116	6759		−1357			
3	167.6	237		70	13230	2.217	5254		−8116	6731		−1385			
4	187.8	231		70	13230	2.485	5740		−8116	6560		−1556			
5	178.2	225		70	13230	2.358	5305		−8116	6390		−1725			
6	35.2	209		70	13230	0.466	974		−8116	5936		−2180			
7	101.4	199		70	13230	1.342	2671		−8116	5652		−2464			
8	101.4	191		70	13230	1.342	2563		−8116	5424		−2692			
9	93.0	194		35	6620	0.616	1195		−1506	5510		+4004			
10	107.6	88		0	0	0	0		5114	2499		+7613			
11	123.2	− 32		0	0	0	0		5114	− 909		+4205			
12	190.8	−169		0	0	0	0		5114	−4800		+ 314			
13	37.8	218		70	13230	0.500	1090		−8116	6191		−1925			
14	37.8	200		70	13230	0.500	1000		−8116	5680		−2436			
15	60.6	182		70	13230	0.802	1460		−8116	5169		−2947			
16	47.2	155		70	13230	0.624	967		−8116	4402		−3714			
17	57.2	128		49.7	9390	0.537	687		−4276	3635		− 641			
18	62.8	73		2.5	473	0.030	22		4641	2073		+6714			
19	213.6	21		0	0	0	0		5114	596		+5710			
20	225.6	−100		0	0	0	0		5114	−2840		+2274			
21	183.8	−196		0	0	0	0		5114	−5566		− 452			
22	148.2	−235		0	0	0	0		5114	−6674		−1560			
23	237.8	−239		0	0	0	0		5114	−6788		−1674			
24	237.8	−241		0	0	0	0		5114	−6844		−1730			
25	227.2	−243		0	0	0	0		5114	−6901		−1787			
26	62.0	−199		0	0	0	0		5114	−5652		− 538			
	3393					17.35×10^6	37.36×10^8						−38.07	−57.15	
	$=A$					$=\sum_7$	$=\sum_8$						+39.24	+57.10	

$A = \Sigma\Delta A = 3393$ in^2 $\quad \sum_7 = 17.35\times10^6$ $\quad \sum_8 = 37.36\times10^8$ $\quad I_{oz} = 1.314\times10^8$ in^4 $\quad \sigma_1 = \alpha ET$

$\sigma_2 = \frac{1}{A}\sum_7 = 5114$ psi $\quad \sigma_3 = \frac{\bar{y}}{I_{oz}}\sum_8 = 28.40\,\bar{y}$ $\quad \sigma_4 = 0$ $\quad$ Checks: $\Sigma\sigma(\Delta A)\approx 0$, $\Sigma\sigma(\Delta A)\bar{y}\approx 0$ $\quad \alpha E = 189$

This may also be written as

$$\sigma = -\alpha ET + \frac{k}{A_\varepsilon}\int \alpha ET\, dA + \frac{ky}{(I_{oz})_\varepsilon}\int \alpha ET\, y\, dA + \frac{kz}{(I_{oy})_\varepsilon}\int \alpha ET\, z\, dA$$

where $k = \left(\frac{E}{E_\varepsilon}\right)$, $A_\varepsilon = \int \frac{E}{E_\varepsilon} dA$, $(I_{oz})_\varepsilon = \int \frac{E}{E_\varepsilon} y^2\, dA$, $(I_{oy})_\varepsilon = \int \frac{E}{E_\varepsilon} z^2\, dA$, and

E_ε is an arbitrary constant and is in our examples taken as E_{steel}.

This expression for σ gives the stress distribution for any given change T in temperature distribution in a nonhomogeneous beam. Equation [2] reduces to Equation [1] when the same value of E applies throughout the beam.

The stress distribution in a composite model box girder has been computed by this method utilizing the same tabular form, Table 1, as used for the homogeneous beam.

The computed stress distribution checks quite well with the experimentally determined stress distribution. The experimental model test data were taken from Reference 3.

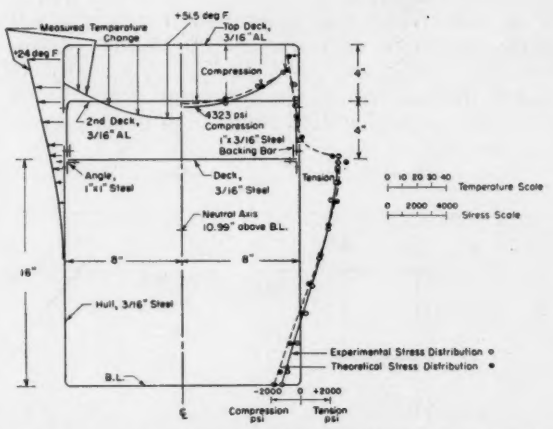


Figure 3. Comparison of Theoretical and Experimental Stress Distribution for a Ship Model.

The experimental data given here are taken from Reference 3.
The model tests were conducted in air.
The distribution of stresses and temperature variations are symmetrical with respect to the centerline of the model.

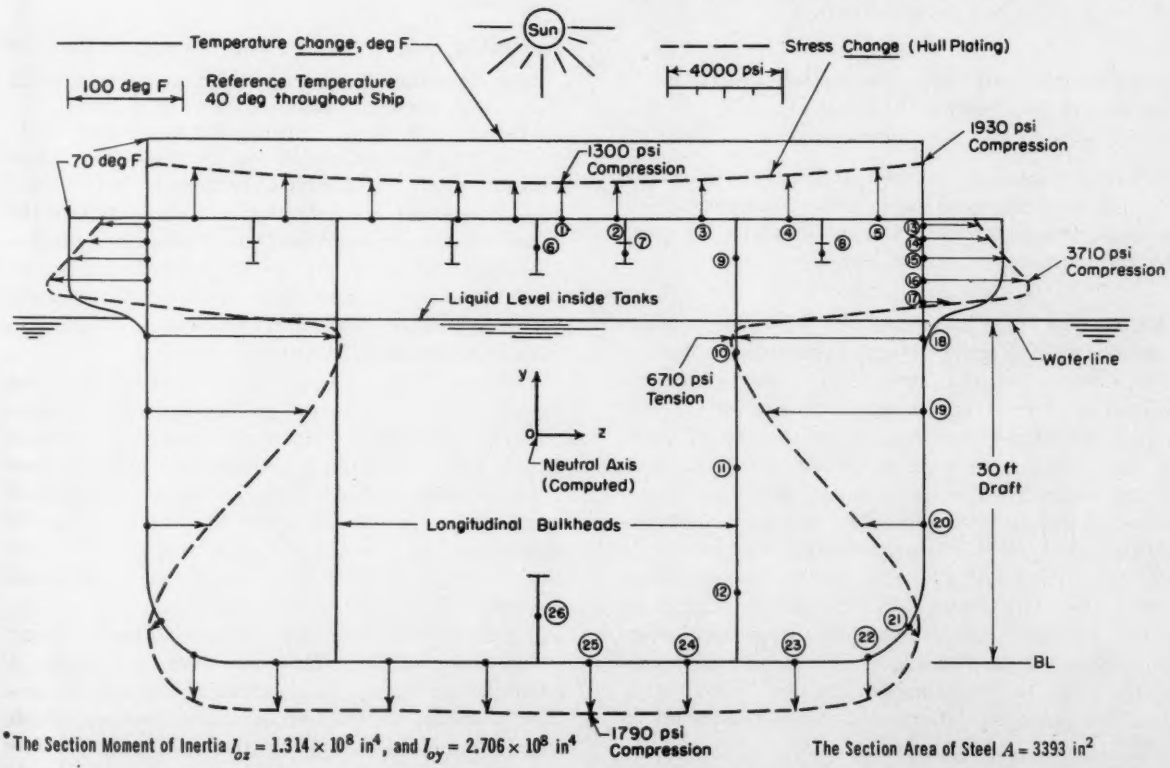

Sun
Temperature Change, deg F
100 deg F
Reference Temperature 40 deg throughout Ship
Stress Change (Hull Plating)
4000 psi
70 deg F
1300 psi Compression
1930 psi Compression
3710 psi Compression
Liquid Level inside Tanks
Waterline
6710 psi Tension
Neutral Axis (Computed)
30 ft Draft
Longitudinal Bulkheads
BL
1790 psi Compression
*The Section Moment of Inertia $I_{oz} = 1.314 \times 10^8$ in^4, and $I_{oy} = 2.706 \times 10^8$ in^4
The Section Area of Steel $A = 3393$ in^2

Figure 4. Theoretical Variation in Longitudinal Hull-Girder Stresses Induced by a Symmetrical Variation in Temperature Distribution.

The circled numbers denote the location of the center of gravity of the respective element of the beam.

BOX BEAMS WITH TRANSVERSE RESTRAINTS

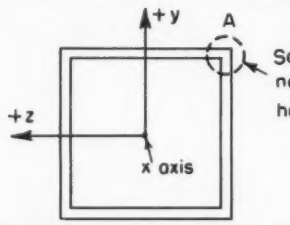
Up to this point it has been assumed that the beam (box beam) is free of transverse restraint. The ship girder, however, does contain transverse constraints in the form of transverse bulkheads which do, to some extent at least, prevent the free transverse expansion of the sections when a temperature change is imposed. It might be of interest, therefore, to examine the effect on the stress distribution if this transverse constraint were complete. Actually, of course, the complete constraint is not attained. The following assumptions, in addition to those utilized thus far, will be made:

1. The walls of the box beam are thin so that there are no stresses normal to the thickness of the wall.

2.* The cross-sectional shape and dimensions are preserved by closely spaced, imaginary diaphragms which are rigid in the plane of the diaphragm and free to warp out of their plane.

3.* The diaphragms are insulated from the box beam and the temperature of the diaphragms remains constant while the walls of the beam are subjected to temperature changes. Let us now determine the stresses σ_1 which, as before, are required to prevent the thermal expansion of all longitudinal fibers of the beam, i.e., $\epsilon_x=0$.

The stress, strain, temperature relationships, are as follows: [4]



$$e_x = \frac{1}{E}[\sigma_x - \nu(\sigma_y + \sigma_z)] + \alpha T, \text{ here } e_x = 0$$

$$e_y = \frac{1}{E}[\sigma_y - \nu(\sigma_x + \sigma_z)] + \alpha T \qquad [3]$$

$$e_z = \frac{1}{E}[\sigma_z - \nu(\sigma_x + \sigma_y)] + \alpha T$$

The nomenclature is given in Figure 2. *For horizontal surfaces*** (parallel to the xz plane, such as the top deck) $e_z=0$ by Assumption 2. From Equations [3] equating e_x to e_z it is found that

$$\sigma_x=\sigma_z$$

Utilizing this result and Assumption 1 (that $\sigma_y=0$) the first of Expressions [3] gives

$$\sigma_x=\sigma_z=-\alpha ET/(1-\nu)$$

Similar reasoning shows that the desired normal stresses σ_1 in the axial direction and the pheripheral normal stresses σ_s parallel to the yz plane are given by the expression derived above:

$$\sigma_1=\sigma_x=\sigma_z=-\alpha ET/(1-\nu)$$

We may therefore substitute this value of σ_1, which includes full effective lateral restraint, in place of the value $\sigma_1=-\alpha ET$ previously used, which assumed no lateral restraint whatever. None of the remainder of the development of the theory given in the preceding sections is affected by the addition of the restraint. The net result is that the magnitudes of the stresses computed under the assumption of no restraint are multiplied by the factor $1/(1-\nu)$ when full transverse restraint is assumed.

It is therefore evident that the stress changes induced by temperature changes are greater when transverse restraint is acting than would otherwise be the case. If, for example, Poisson's ratio $\nu=1/3$ then the stress variations will be increased by 50 percent. The actual stresses in a box beam would be expected to lie somewhere between those obtained for the assumption of zero restraint and full transverse restraint.

PROCEDURE FOR COMPUTING THE STRESS DISTRIBUTION

It is convenient to arrange the necessary data and numerical work in tabular form as in Table 1.

A plan of the ship's midship section should be obtained. On this plan plot the distribution of temperature changes for which the changes in stress are to be computed. Next divide the cross section of the beam into a convenient number of area elements ΔA. In regions of large changes in temperature gradients, for example near the waterline, a finer subdivision should be made than elsewhere in order to define the stress distribution better.

The area of each element ΔA, its position from the principal centroidal axes, and the average temperature change T of the element are entered in Columns 2, 3, 4, and 5 of Table 1. Columns 6, 7, 8, and 9 are filled in next; the summation of these columns will give the values of the integrals which appear in Equations [1] and [2]. Columns 9 through 13 are then filled in. Column 13 gives the required stress distribution† as defined by Equations [1] or [2]. Columns 14 and 15 provide a check on the accuracy of the computation. The summation of Column 14 gives the net change in longitudinal force at the section, incident to the temperature changes which should equal zero; similarly the summation of Column 15, representing the resulting change in bending moment or couple at the section, should also equal zero. Due to the approximations and inaccu-

*This type of imaginary constraint has been used by other authors. Other types of constraints could be thought of which would serve the same purpose and give the same result.

**Note that this solution is not applicable at the corners of the box.

†If transverse restraint exists, the stress variations will be larger as discussed above.

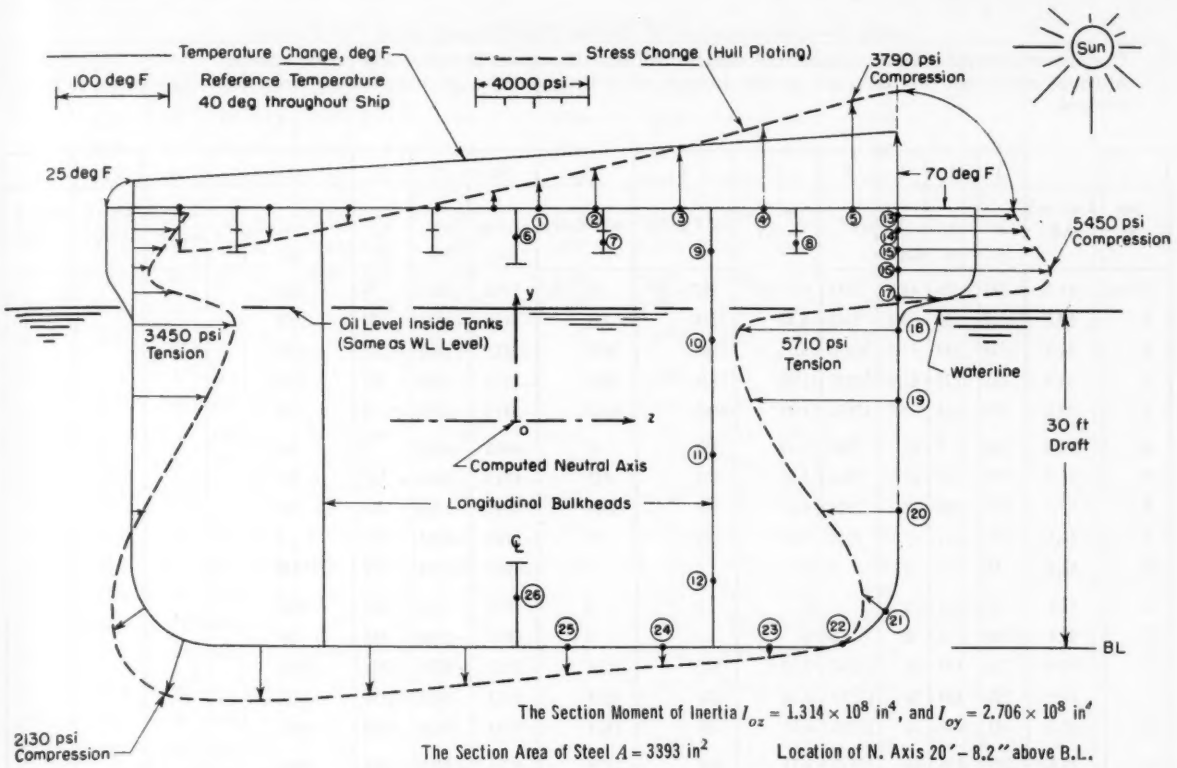


Figure 5. Theoretical Variation in Longitudinal Hull-Girder Stresses Induced by a Non-Symmetrical Variation in Temperature Distribution.

The circled numbers denote the location of the center of gravity of the respective element of the beam.

racies of numerical calculations one can only hope to approximate the requirements of zero moment and force.

DISCUSSION

Application of the theory to a test of a model ship indicates fairly good agreement between computed and measured stresses. No full-scale data on the distribution of temperature variations and corresponding stress variations are available to the author for checking the applicability of this method to full-scale ships. It would seem reasonable to assume however, that this theory (which assumes that the variation in temperature distribution is the same in every transverse section) would be valid to the same degree that ordinary beam theory is applicable.

The computations given here for a T-2 oil tanker indicate that rather severe stress gradients may occur in the side shell plating due to temperature changes. It is furthermore evident that the temperature-induced stresses may be appreciably larger in the side plating (near the waterline) than in the deck plating. The assumed conditions for the T-2 tanker calculations, as well as for the model experiment, involved heating of the upper deck which induced mostly compressive stresses in the deck and bottom. It could equally well have been that the temperature of the air, and consequently of the exposed portion of the ship, had fallen below the water temperature, as in freezing weather. Inspection of Equation [2] will show that in this case the stress distribution would be identical except that compressive stresses would now be tensile stresses and vice versa, assuming the magnitudes of the temperature changes were identical.

Acker [2] has stated that "when high diurnal thermal stresses develop (stress changes) they are usually compressive and should cause no serious trouble." This statement would not appear to be justified if one considers the stresses which would be induced by low air temperatures and relatively high water temperatures, such as would be expected during freezing. Even under the more usual conditions in which the deck is heated, there will be, according to the theory, relatively large tensile stresses induced in the shell plating near the waterline; see Figures 4 and 5. One interesting observation is that stresses of the same sign may be induced in *both the deck and the bottom* of the ship by the type of temperature changes expected in service; see Figure 4. Thus a cooling of the above water portion of the hull would induce tensile stresses in both the deck and

TABLE 3—*Computation of Stress Distribution in a T-2 Tanker*

Unsymmetric distribution of temperature changes. All units are given in inches and pound system. Although more than two digits are carried through in the computations, no more than two significant figures are to be inferred.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Item No	Area, in^2 ΔA	$\bar{y}$ in.	$\bar{z}$ in.	T deg F	αET	$\alpha ET \times \Delta A$	$\alpha ET(\Delta A)\bar{y}$	$\alpha ET(\Delta A)\bar{z}$	$\sigma_1 + \sigma_2$ psi	σ_3 psi	σ_4 psi	$\sigma = \sigma_1 + \sigma_2 + \sigma_3 + \sigma_4$ psi	$\Sigma F = 0$ $\sigma(\Delta A)$	$\Sigma M = 0$ $\sigma(\Delta A)\bar{y}$	$\Sigma M = 0$ $\sigma(\Delta A)\bar{z}$
1 Stbd	44.5	240	24	48.9	9240	4.11×10^5	987×10^5	97×10^5	−5743	+4661	+ 76	−1010			
2	89.0	238	93	52.8	9980	8.88	2114	826	−6483	+4622	+ 296	−1570			
3	83.8	237	183	57.8	10920	9.15	2169	1675	−7423	+4603	+ 582	−2240			
4	93.9	231	275	62.8	11870	11.15	2575	3065	−8373	+4486	+ 875	−3010			
5	89.1	225	364	67.8	12810	11.41	2568	4155	−9313	+4370	+1158	−3790			
6	17.6	209	0	42	7940	1.40	292	0	−4443	+4059	0	− 380			
7	50.7	199	120	42	7940	4.03	801	483	−4443	+3865	+ 382	− 200			
8	50.7	191	300	42	7940	4.03	769	1208	−4443	+3709	+ 954	+ 220			
9	46.5	194	211	42	7940	3.69	716	779	−4443	+3767	+ 671	0			
10	53.8	88	212	0	0	0	0	0	+3497	+1709	+ 674	+5880			
11	61.6	− 32	214	0	0	0	0	0	+3497	− 621	+ 681	+3560			
12	95.4	−169	215	0	0	0	0	0	+3497	−3282	+ 684	+ 900			
13	18.9	218	407	70	13230	2.50	545	1018	−9733	+4234	+1294	−4210			
14	18.9	200	407	70	13230	2.50	500	1018	−9133	+3884	+1294	−4560			
15	30.3	182	405	70	13230	4.01	730	1624	−9733	+3534	+1288	−4911			
16	23.6	155	405	70	13230	3.12	484	1624	−9733	+3010	+1288	−5440			
17	28.6	128	405	49.7	9370	2.69		1088	−5893	+2486	+1288	−2120			
18	31.4	73	405	2.5	473	0.15	11	60	+3024	+1418	+1288	+5730			
19	106.8	21	405	0	0	0	0	0	+3497	+ 408	+1288	+5190			
20	112.8	−100	405	0	0	0	0	0	+3497	−1942	+1288	+2840			
21	91.9	−196	396	0	0	0	0	0	+3497	−3806	+1259	+ 950			
22	74.1	−235	348	0	0	0	0	0	+3497	−4564	+1107	+ 40			
23	118.9	−239	265	0	0	0	0	0	+3497	−4641	+ 843	− 300			
24	118.9	−241	154	0	0	0	0	0	+3497	−4680	+ 490	− 690			
25	113.6	−243	54	0	0	0	0	0	+3497	−4719	+ 172	−1050			
26 Stbd	31.0	−199	0	0	0	0	0	0	+3497	−3865	0	− 370			

the bottom structure. This could be significant from the standpoint of brittle fractures.

The temperature-induced stresses computed for the T-2 tanker by means of the theory (neglecting transverse restraints, if any) appear to be low when compared with the measured stress variations on such a vessel (Figure 1) where a variation of 7000 psi on the main deck is indicated. Presumably one could have measured even greater stresses in the side plating. It may be that the temperature changes assumed for the longitudinal deck beams were too low, that transverse bulkheads provided appreciable transverse restraints or that there existed an appreciable nonuniformity in the longitudinal temperature distribution. The only convincing way of checking the validity of the theory, as applied to ships, is to obtain simultaneous measurements of temperature and stress variations on one or more typical ships. It is *strongly recommended* that such full-scale tests be made. It is also suggested that model tests of box girders, with and without transverse restraint, with and without changes in temperature gradients in the longitudinal direction may be worthwhile in evaluating the theory.

Meanwhile it appears reasonable to assume that the temperature-induced stresses need to be considered in the strength design of the hull girder.

ACKNOWLEDGMENTS

The author's interest in this problem was aroused as a consequence of occasionally fairly large diurnal stress variations which were measured on the *Esso Asheville* in connection with a program of determining service stresses in full-scale hull structures which is being carried out under the guidance of the

TABLE 3—*Continued*

1 Port	44.5	+240	− 24	46.2	8730	3.89	932	− 93	−5233	+4661	− 76	− 650				
2	89.0	238	− 93	42.4	8010	7.13	1697	− 663	−4513	+4622	− 296	− 190				
3	83.8	237	−183	37.4	7070	5.93	1404	−1084	−3573	+4603	− 582	+ 450				
4	93.9	231	−275	32.4	6120	5.75	1328	−1580	−2623	+4486	− 875	+ 900				
5	89.1	225	−364	27.4	5180	4.62	1038	−1680	−1683	+4370	−1158	+1600				
6	17.6	209	0	42	7940	1.40	292	0	−4443	+4059	0	− 384				
7	50.7	199	−120	42	7940	4.03	801	− 483	−4443	+3865	− 382	− 960				
8	50.7	191	−300	42	7940	4.03	769	−1208	−4443	+3709	− 954	−1690				
9	46.5	194	−211	42	7940	3.69	716	− 779	−4443	+3767	− 671	−1350				
10	53.8	88	−212	0	0	0	0	0	+3497	+1709	− 674	+4530				
11	61.6	− 32	−214	0	0	0	0	0	+3497	− 621	− 681	+2200				
12	95.4	−169	−215	0	0	0	0	0	+3497	−3282	− 684	− 470				
13	18.9	218	−407	25	4730	0.89	195	− 364	−1233	+4234	−1294	+1710				
14	18.9	200	−407	25	4730	0.89	179	− 364	−1233	+3884	−1294	+1360				
15	30.3	182	−405	25	4730	1.43	261	− 580	−1233	+3534	−1288	+1010				
16	23.6	155	−405	25	4730	1.12	173	− 452	−1233	+3010	−1288	+ 490				
17	28.6	128	−405	18.1	3420	0.98	125	− 396	+ 77	+2486	−1288	+1280				
18	31.4	73	−405	1.3	246	0.08	6	− 31.3	+3251	+1418	−1288	+3380				
19	106.8	21	−405	0	0	0	0	0	+3497	+ 408	−1288	+2620				
20	112.8	−100	−405	0	0	0	0	0	+3497	−1942	−1288	+ 270				
21	91.9	−196	−396	0	0	0	0	0	+3497	−3806	−1259	−1570				
22	74.1	−235	−348	0	0	0	0	0	+3497	−4564	−1107	−2170				
23	118.9	−239	−265	0	0	0	0	0	+3497	−4641	− 843	−1990				
24	118.9	−241	−154	0	0	0	0	0	+3497	−4680	− 490	−1670				
25	113.6	−243	− 54	0	0	0	0	0	+3497	−4719	− 172	−1390				
26 Port	31.0	−199	0	0	0	0	0	0	+3497	−3865	0	− 370				
	$A=\Sigma\Delta A$ =3393 in^2					Σ_7=118.65 ×10^5	Σ_8=25521 ×10^5	Σ_9=8605 ×10^5					−29.66 +29.77	−45.92 +44.57	−85.0 +87.3

$I_{oy}=2.706\times10^8$

$I_{oz}=1.314\times10^8$

$\alpha E=189$

$\sigma_1=-\alpha ET$ $\quad \sigma_2=\frac{1}{A}\Sigma_7=\frac{118.65\times10^5}{3393}=+3497$

$\sigma_3=\frac{y}{I_{oz}}\Sigma_8=19.42\bar{y}$ $\quad \sigma_4=\frac{z}{I_{oy}}\Sigma_9=3.180\bar{z}$

Checks: $\Sigma\sigma\Delta A=0$

$\Sigma\sigma\Delta A\bar{y}=0$

$\Sigma\sigma\Delta A\bar{z}=0$

S-10 panel of the SNAME. The possible importance of such temperature-induced stresses in connection with hull fractures was impressed on the author by Mr. Arthur R. Gatewood of the American Bureau of Shipping. This work is part of a broad study of the strength requirements of ships being carried out at the David Taylor Model Basin under the cognizance of the Bureau of Ships, Code 440. The encouragement received from Captain E. A. Wright, formerly at the Bureau of Ships, and Mr. J. Vasta of the Bureau of Ships has made this study possible.

The theoretical development has been checked by Mr. J. E. Greenspon and the numerical calculations were made by Mr. R. R. Milam, both of the Taylor Model Basin staff. Special thanks are due the Esso Company for making a tanker available for the use of the S-10 panel; without the background of these full-scale tests this particular study would probably not have been undertaken at this time. The constructive comments of the members of the S-10 panel and of Mr. Gatewood and Mr. H. G. Acker on a preprint of this report were greatly appreciated. This report was critically reviewed by Mr. R. T. McGoldrick and Dr. E. H. Kennard of the Taylor Model Basin staff. Dr. Kennard made helpful suggestions which simplified the treatment of the nonhomogeneous beam. Thanks are due the Institution of Naval Architects for permission to use Figure 16 of Reference 4, which made it possible to have one experimental check of the theory.

APPENDIX

MEASUREMENT OF STRESSES INDUCED BY TEMPERATURE VARIATION

The question frequently arises as to how one can

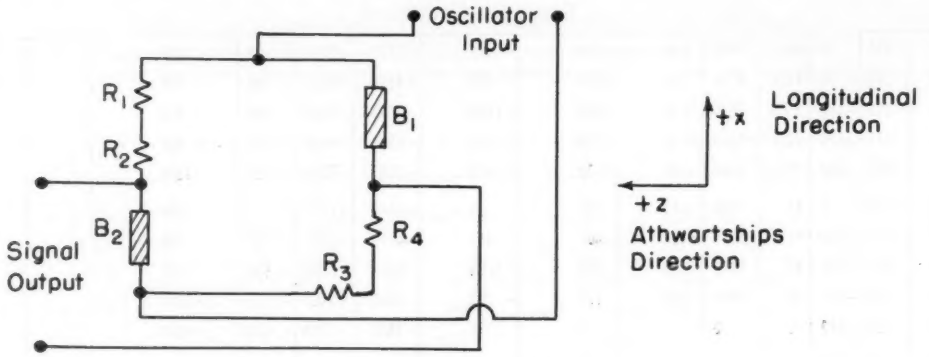


Figure 6. Circuit for Measuring σ_x.

measure the actual stress variation in a structure including those stress variations which are a consequence of a variation in temperature distribution throughout the structure. Inasmuch as the stresses induced by changes in temperature arise due to restraints which prevent the free expansion of the material, it is sometimes stated that the temperature-induced stresses are "strainless" stresses.

The relationship between stresses, strains, and temperature changes, as given in article 65 of Reference 4, have been given by Equation [3]. These relationships specify the actual strains corresponding to any actual change in stress and the corresponding change in temperature. It is immaterial whether the stresses are due to temperature changes alone, or to external loading, or to a combination of both—these relationships are valid in either case.

For the stresses in the plating of a ship girder it may be assumed that the stress σ_z normal to the thickness of the plating is zero.

Utilizing the expressions for e_x and e_z, Equations [3], the stress σ_z may be eliminated between them, and the following expression for σ_x results:

$$\sigma_x = \frac{E}{1-\nu^2}[e_x + \nu e_z - (1+\nu)\alpha T] \qquad [4]$$

where σ_x is the change in normal stress in the longitudinal direction,
e_x, e_z are the total changes in strains in longitudinal and transverse directions, respectively,
ν is Poisson's ratio,
α is the temperature coefficient of expansion, and
T is the change in temperature at the point of measurement.

If there are no variations in external loads, then the stress σ_x and the strains e_x, e_z will be due only to the changes in temperature throughout the structure, including the temperature change T at the point x, y, z at which σ_x and e_x, e_z are determined.

To determine σ_x it is then only necessary to measure the quantities e_x, e_z, and T, assuming that E, ν, and α are known constants, and substitute these values in Equation [4].

Another, more convenient method is to arrange an electrical circuit which performs the operation indicated by the right-hand side of Equation [4], thus yielding a signal which is proportional to the actual stress variation in the structure. This is the procedure which was adopted for the stress measurements on the T-2 tanker *Esso Asheville*. The circuitry is shown in Figure 6, and its operation is as follows. R_1, R_2, R_3, and R_4 are 60-ohm SR4 strain gages mounted on the deck structure. B_1 and B_2 are 120-ohm SR4 strain gages mounted on steel bars which are free to expand and which are at the same temperature as is the deck. Bars B are physically located next to resistors R_1, R_2, R_3, and R_4. These gages are all situated close together at the location at which the stress is measured.

The resistance change of an SR4 strain gage is proportional to the strain of the metal to which it is attached and also proportional to the resistance of the gage.

Consider Equation [4] term by term and consider the component of the circuit which provides an electrical output proportional to that term.

1. A signal proportional to $+e_x$ is provided by 180 ohms of strain gage wire $(R_1+R_2+R_4)$ physically oriented in the longitudinal x direction. Elementary electrical theory shows that the voltage changes across R_1, R_2, and R_4 are additive at the output terminals.

2. A signal proportional to $+\nu e_z$ is provided by ν times 180 ohms of strain gage wire (R_3) physically oriented in the athwartships z direction. Taking $\nu = 1/3$, it is evident that $1/3 \times 180 = 60$ ohms are required for R_3. The signal output of gage R_3 is electrically additive to that of gages R_1, R_2, and R_4.

3. A signal proportional to $-(1+\nu)\alpha T = -4/3\alpha T$ is provided by $4/3 \times 180 = 240$ ohms of strain gage wiring attached to steel bars which are free to elongate an amount αT per unit of length where αT is the temperature induced "stress-less" strain. The electrical output of gages B_1 and B_2 is electrically

subtracted from the output of R_1, R_2, R_3, and R_4.

It should be noted that any change of resistance of the strain gage wire itself, as a consequence of temperature changes, is automatically compensated by having the same amount of strain gage wire, 120 ohms in this case, in each arm of the bridge.

It has been shown, therefore, that the electrical output of the strain gage bridge is proportional to the stress σ_x at the surface of the structure (i.e., at the location of the gage).

REFERENCES

1. Osgood, W. R., "Residual Stresses in Metals and Metal Construction," Reinhold Publishing Company, New York (1954).
2. Acker, H. G., "Highlights of Welded Ship Research," Transactions Society of Naval Architects and Marine Engineers (1949), Vol. 57.
3. Corlett, M. A., "Thermal Expansion in Composite Ships," Transactions Institute of Naval Architects (1950), Vol. 92.
4. Timoshenko, S., "Theory of Elasticity," McGraw-Hill Book Company, Inc., New York (1934).
5. Mar, J. W., and Engel, S. J., "Experimental and Analytical Determination of the Transient Thermal Stresses in a One Cell Box Beam," Aerolastic and Structures Research Laboratory, Massachusetts Institute of Technology Technical Report 25-13 (May 1954).
6. Melan, E., and Parkus, H., "Wärmespannungen infolge stationärer Temperaturfelder," Springer Verlag, Wien (1953). This book has an excellent bibliography on the subject of thermal stresses.

Courtesy of New York Shipbuilding Corp.

U.S.S. BOSTON (CAG1) the Navy's first guided missile cruiser as she left the builders' yard on completion.

HOW MECHANICAL ENGINEERS ARE EDUCATED IN RUSSIA

ACKNOWLEDGEMENT

B. L. Goodlet, Deputy Chief Engineer of the Atomic Research Establishment, Harwell, England, is the author of this article which was published in the February 10, 1956, edition of "Engineering."

IN NOVEMBER, 1955, a party of atomic scientists visited Moscow as guests of the Academy of Sciences of the U.S.S.R. I was one of this party and asked for a visit to an institute training mechanical engineers to be included in my program. (It should be noted that I am familiar with the Russian language.) The present article sets out the information I collected and my impressions.

MONOTECHNICAL INSTITUTES

The Moscow Higher Technical Institute (M.V.-T.U.), named after N. E. Baumann, was founded as a trade school in 1830, converted into a higher technical institute in 1868, enlarged under new statutes in the nineties and, after the revolution of 1917, expanded into an institute of the polytechnic type.

In 1932 the Russian Government decided to abolish polytechnics and educate specialists in monotechnical institutes having close ties with the appropriate branch of industry. In conformity with this decision the M.V.T.U. faculties of electrotechnics, aviation, structural engineering and chemical technology broke away as independent institutes on other sites, and the M.V.T.U. was left with the basic job of turning out mechanical engineers for the heavy machine industry. The budget of the school just after this change was 8.3 million rubles per annum (about £0.83 million sterling), while in 1937 26 million rubles were voted for extensions. The output in 1934 was 450 graduates. During the war the Institute was evacuated from Moscow but without serious interruption of its work. Since the war there has been steady expansion and at the present time there are about 10,500 full-time students and a teaching staff of 700. In 1955 the Institute celebrated the 125th anniversary of its foundation and was awarded the Order of Lenin for its long and excellent record of service.

The principal faculties and constituent chairs existing in the Institute at the present time are listed in Table I. There may be some minor inaccuracies regarding the distribution of chairs among faculties, but the list is otherwise a fair indication of the organization.

TABLE I.—*Faculties and Chairs, the Moscow Higher Technical Institute.*

I. *Faculty of Thermal and Hydraulic Machines*
- Chair of Boiler Engineering
- Chair of Steam Engines
- Chair of Steam Turbines
- Chair of Gas Turbines—Land and Mobile
- Chair of Gas Turbines—Aviation
- Chair of Internal Combustion Engines
- Chair of Refrigerating Machines and Compressors
- Chair of Low-Temperature Work
- Chair of Hydraulic Machines and Turbines

II. *Faculty of Machine Construction*
- Chair of Strength of Materials
- Chair of the Theory of Mechanisms and Machines
- Chair of Machine Details
- Chair of Hoisting and Conveying Machinery

III. *Faculty of Precision Instruments*
- Chair of Horology
- Chair of Calculating Machines
- Chair of Precision-Instrument Design
- Chair of Optical Instruments
- Chair of Metrology
- Chair of Special Instruments
- Chair of Precision-Instrument Technology

IV. *Faculty of Transport Machines*
- Chair of Locomotive Engineering
- Chair of Automobile Engineering

V. *Faculty of Mechanical Technology*
- Chair of the Technology of Machine Construction
- Chair of Machine Tools
- Chair of Welding and Flame Cutting
- Chair of Pressure Working of Metals

VI. *Faculty of Metal Technology*
- Chair of Metallurgy and Heat Treatment
- Chair of Foundry Technology

HIGH DEGREE OF SPECIALIZATION

The whole course of study totals about 4,870 hours of instruction spread over 5½ years of full-time attendance, and the qualification awarded is that of Mechanical Engineer. The general plan is that all students take the same courses in the first three years, have some common courses in the fourth year, and during the fifth and sixth years concentrate on some specialized branch of mechanical engineering; the last half year is spent on a design project in that branch. A list of some of the 23 specializations available is given in Table II.

TABLE II.—*Subjects in which Students Specialize.*

Boiler engineering
Turbine engineering—steam
Turbine engineering—gas
Turbine engineering—aviation
Refrigerating engineering—refrigeration plant
Refrigerating engineering—compressor design
Refrigerating engineering—low temperature work
Hydraulic turbines and hydraulic machines
Internal-combustion engines
Optical-mechanical instruments
Precision instruments—horology
Precision instruments—metrology
Thermo-technical measuring instruments and controllers
Hoisting and conveying machinery
Transport Machinery—locomotives
Transport Machinery—automobiles
Welding and flame cutting
Foundry technology
Pressure working of metals
Metallurgy and heat treatment

This list shows that the degree of specialization required of a Russian mechanical enigneer at graduation is much greater than would be regarded as proper at any British university. It should, however, be noted that this specialization is achieved by a longer period of study and does not imply that insufficient time is spent on the teaching of fundamentals. The common courses taken by all students occupy some two-thirds of the total teaching time, and the time spent on basic subjects is as great as in British university courses. This will be clear from the approximate time schedule of the common courses which appears in Table III.

The total time spent on Groups A. B. and C is some 2,740 hours. Group D includes subjects of considerable educational value often missing in British engineering curricula. Group E may be ridiculed as political indoctrination, which it undoubtedly is; it might however be more profitable to think of it as the Russian equivalent to a set of lectures on the British Constitution and the working of our Government machine, which would undoubtedly be regarded as a highly educative addition to the average British engineering syllabus.

DIPLOMA PROJECT

The culmination of the course is the "diploma project," which occupies some 20 weeks of the final year. The problem presented is usually the design of an engine, crane or other machine appropriate to the student's specialization, and this has to be worked out in great detail leaving nothing to the imagination. I looked at several diploma-project reports and was surprised at the excellence of the draftsmanship. One of these reports dealt with the design of a motor cycle with fluid transmission and I was particularly impressed with the care taken to work out and specify all the operations of its manufacture, over and above the many problems of the design itself. We were told that before starting his diploma project a student spent a few weeks in a factory which made the kind of machine he would be expected to design.

TABLE III.—*Schedule of Common Courses*

Subject	Total hours	Group hours
Group A		
Mathematics	416	
Physics	245	
Chemistry	120	
Theoretical mechanics	215	
Electricity	162	
Applied heat	90	
	—	1,248
Group B		
Geometry	90	
Drawing and machine drawing	188	
Details of machines	215	
Theory of machines	197	
Strength of materials	232	
	—	922
Group C		
Technology of metals	182	
Metallurgy and heat treatment	96	
Instructional workshops	204	
Workshop measurements and interchangeable manufacture	48	
Industrial safety and fire drill	40	
	—	570
Group D		
Political economy	140	
Industrial economics, organization and planning	95	
History of technology	39	
Foreign language	134	
	—	408
Group E		
Fundamentals of Marx-Leninism	252	252

The great importance attached to knowledge of workshop processes and machine-tool operations was apparent both in the diploma reports and in the workshops and laboratories, the equipment of which was excellent.

ENTRY STANDARDS AND GRANTS

We asked how entrants were selected, and were told that admission was by competitive examination at the age of 17-18; there were always several times more applicants than vacancies. Admission was easiest to the specializations with a pronounced workshop flavor—for example, forging and pressure working of metals; "white collar courses" such as turbine design were always over popular. I was told that the calculus was not taught in the schools, which put their entry standard probably a year behind that of our grammar-school leavers. However, students have to work extremely hard and there is a lot of homework. Students who fail a course examination are al-

lowed to try again after the vacation but are rejected on failing twice.

Admission to a higher technical institute is a qualification which exempts the successful candidate from military service. The physical education and sports which for students replaces military service occupies only 134 hours in 5½ years. We asked where the armed forces obtained their specialists from and were told that they were trained in special service schools.

Students at the M.V.T.U. receive a State stipend of 300 to 500 rubles per month. In cases of exceptional merit the stipend can go up to 600 rubles per month. The precise sum is fixed by the Principal, having regard to: *(a)* the year which the student has reached; *(b)* the specialization (the rougher options such as pressure working of metals carry higher stipends); and *(c)* the quality of the student's work. All stipends are awarded without regard to parents' income and social origin. Really outstanding men are eligible for one of the 100 Stalin Premia, valued at 800 rubles per month, in the gift of the Principal. Cases of hardship can be helped from a special fund also controlled by the Principal. Students are expected to pay fees which vary between 200 and 400 rubles per annum, but exemption can be obtained by making good progress. The value of the ruble can be roughly fixed from the statement that the wage of a skilled workman in Moscow would be approximately 900 rubles per month. It is, however, improbable that students living in hostels pay the full value of their maintenance.

ADVANCED STUDY AND RESEARCH

We gathered that graduates were expected to spend three years in the industry of their specialization. After this they could if they wished embark on a research career by applying for admission as aspirant at one of the many teaching or research institutes. The higher degree of Candidate in Science (perhaps equivalent to Ph.D.) is open to aspirants of three-years standing. The standard of the ultimate degrees of Doctor of Science or Doctor of Technical Science seems to be very high. The present number of aspirants at the M.V.T.U. is around 200; about one-third of these are brilliant students who were admitted as aspirants directly after graduation and without working in industry.

Each faculty has its own research program, which is planned in close collaboration with the factories with which the faculty is linked. To the writer the most interesting and significant feature was the amount of research in progress in the fields of theory of mechanisms and machines, design of machine details, machine tools, and technology of machine construction. One has the feeling that in Russia science has closer links with drawing office and workshop practice than is the case in England.

Another point of interest is the large number of excellent, up-to-date and advanced text-books available to Russian engineers. The writing of such text-books is listed among the important research achievements of the staff, showing the great importance attached to good teaching.

SCIENTIFIC APPROACH TO WORKSHOP

The present set-up of the M.V.T.U. is the product of history and Communist philosophy. It is an accident of history that most continental universities refused to admit the ignoble art of engineering to their disciplines so that special higher technical institutes had to be set up for the training of engineers. These new technical universities had to set high standards in science in order to compete in standing and prestige with the older foundations. However, not having to conciliate the arts faculty, they could also teach real engineering—not merely bastard science.

In England, where the art of engineering was pioneered by craftsmen, the need for teaching workshop technology was hardly felt; the demand was for simple science to aid the practical engineer.

In Russia, however, industrialization was imposed from above; there was no large pool of craft skill, and workshop technology could not be learned from peasants. Russian engineering curricula have therefore always included a much larger amount of instruction in machine design and workshop technology than has been hitherto thought necessary in England. We have, however, now entered a new era in which workshop methods are no longer empirical, but are the result of scientific and operational research. In this new era workpeople are no longer reliable teachers for our student engineers and the traditional Russian policy of giving formal instruction in machine tools, workshop technology and production methods is likely to be the key to the future.

TECHNOLOGY UNDER COMMUNISM

Communism is defined in my dictionary as a doctrine advocating the common ownership of wealth. However, wealth has to be produced before it can be distributed and practical Communism seems to be much more concerned with production. Nowadays, wealth is produced by means of machines, which therefore occupy a special position of honor in Communist philosophy. The machine which produces wealth and lightens human toil, the machine tool which produces the machines, the prime mover which supplies power, electricity which transmits and applies power, the transport system which moves produce to where it is needed, these are the pillars of the Communist system. It is, therefore, hardly surprising that the design, construction and technology of machines should occupy a prominent place in the Russian engineering curriculum. Moreover the Communist prophets regarded science as the ultimate wisdom and used "empirical" as a term of abuse. It is therefore not surprising that the Russian approach to workshop technology should be strictly scientific.

Opinions on the desirability of educating engineers in monotechnic schools will undoubtedly differ. The writer's view is that the Russian syllabus of common lectures is as broad as that of most British university engineering schools and, provided, these lectures are not given by people with a monotechnic outlook, the fact that the listeners have a common interest will hardly matter. The question of size is also very rele-

vant—if one wishes to teach very large numbers (10,000 at the M.V.T.U.) the monotechnic institute probably has a lot to recommend it.

Russian engineering curricula are clearly designed to convey a lot of useful information. Probably they have to be of this pattern in order to satisfy the Minister of Higher Education who must approve them. One wonders if this Minister realizes the distinction between what is taught and what is absorbed and whether he knows that students sometimes learn more when they are taught less. With organized instruction filling up to 36 hours per week and "plenty of homework," one wonders when students find time to do anything they are not told to do, and whether their originality and initiative does not atrophy under this discipline.

Certainly Russian scientists, as distinct from engineers, sense a touch of the "conveyor belt" about their curricula and wonder whether our own less overloaded honors courses may not be a better foundation for a research career. There does not, however, seem to be much wrong with Russian research, so their good men must get through in spite of defects in their schooling. If one assumes that good men cannot be spoiled, there is something to be said too for designing a curriculum to cater for the much larger number of average men who learn nothing they are not taught, but who can be taught to be useful engineers. Is it not permissible to suggest that the real object of education is to enable one to function effectively at least one grade above one's natural intellectual level?

GROUNDING IN FUNDAMENTALS

To summarize, I have no reason to think that the Russian graduate engineer is less well grounded in fundamentals than our men. I am certain that he gets a better grounding in engineering drawing, design and manufacturing technology, and is therefore useful to his employer with less further training than our men need. The study of some particular branch of mechanical engineering is, however, largely wasted time unless the graduate does in fact find employment in that branch—and this would be much more difficult to guarantee in England than in Russia.

The production of the cadres of technical specialists at an expanding rate is a cardinal point of Communist doctrine. The present rate of production of engineers from the Russian higher technical institutes may now be around 60,000 a year. The best figures I have been able to discover are given in Table IV.

TABLE IV.—*Numbers of Engineers Trained in Various Periods*

Period	Output of Engineers	
	Total	Yearly Average
1938-41 (3½ years)	116,500	33,500
1941-45 (4½ years)	41,600	9,300
1945-50 (5 years)	170,000	34,000
1951-55 (5 years)	160,000	52,000

The term "engineer" in Russia (and generally on the Continent) implies a man with a qualification of degree standard conferred by a higher technical institute, usually demanding 5½ years study. Since some of the large factories in Russia have technical schools affiliated to the nearest higher technical institute it is probable that some of the engineers listed above graduated by part-time day and evening studies, but I do not think that this fraction is large. The training of technicians is another matter, to be discussed later. On the matter of the rate of production of men trained for research, I have seen it stated that the fourth five-year plan ending in 1950 produced 15,500 young scientific workers trained via the aspirantura and that the fifth five-year plan aimed at doubling this number.

The technician is the warrant officer of industry and occupies an honorable position in the hierarchy without any pretence of being a fully qualified engineer. The various kinds of technician and the range of knowledge and skills each must possess are clearly defined, and special text-books are written to assist each kind to qualify.

TRAINING TECHNICIANS

The training of technicians is carried out *(a)* in technicums and *(b)* in night schools. (No school for training technicians was visited.) The technicum is a day school giving specialized courses of up to three years duration with normal entry at age 16 to 18, after seven or ten years schooling. There are also, however, evening departments catering for working people. Each technicum is associated with a particular group of industries; for example, ferrous metallurgy, machine-tool construction, precision instruments, etc. Some of the specializations noted were: technician geophysicist, technician metallurgical furnace expert, technician wire rope maker, technician electroplater, technician metal cutting expert, technician electrical tester, and technician steel melter.

Equipment for practical instruction is stated to be good. The passing-out standards of technicians are controlled by the State Commission on Qualifications (which passes comments on individual students) and there is no reason to believe that standards in technicums are not high.

It is, however, clear that large numbers of "specialists" are also produced by evening studies. I noted a statement that in 1954 there were 3 million evening students including those studying farming specialities. I also read an account of one workman who, starting as an electrician's mate, had obtained a qualification by evening study, had risen to be chief mechanic of a large works, and was preparing to defend a dissertation for the degree of Candidate of Technical Science. It appears, however, that this degree of self-advancement is unusual.

CRAFT TRAINING

I am uncertain how the time-honored method of apprenticeship operates in Russia, if at all. I suspect it has been replaced by intensive training in trade schools attached to various works. One article de-

scribes a well-equipped trade school giving intensive one-year courses for turners, assembly fitters, repair fitters, electricians, metallurgical-laboratory workers and draftsmen. Young people of both sexes who pass these courses apparently then go straight on to production under the supervision of more experienced workers. The Works League of Young Communists (Komsomol) were told to arrange a recruiting party for school leavers at which the chief constructor and others set out the advantages of working at a trade.

It is interesting to note that everything possible is done to emphasize the dignity of manual work and individuals are referred to as "the distinguished turn-innovator Comrade—" or "the distinguished furnace repair fitter the Stalin Laureate Comrade—" Another feature noted at all institutes is the "Notice Board of Honor" which displays for six months the photographs of workers commended for some special achievement or general good work. Information about innovations and praise for the innovators is disseminated from the "House of Scientific-Technical Propaganda" in Leningrad. There also seems to be a special output of books for young people describing, in the guise of a story, the attractions of work in industries which happen to be short of labor. It is also worth noting that there are at least seven gradations of pay between ordinary and top-grade workmen, based on skill and achievement. This all seems to be pretty good psychology and we can perhaps learn something from the Russians in this as in other fields.

ERNEST BOGHOSIAN

HISTORY OF NAVAL SEARCHLIGHTS

THE AUTHOR

graduated from the University of Pennsylvania in 1941 with the degree of B.S. in E.E. Following graduation he was employed in the Electrical Design Section of the Philadelphia Naval Shipyard. In December 1945 he was transferred to the Electrical Branch of the Bureau of Ships. In addition to other duties he is the illumination and searchlight specialist for the Bureau and has prepared papers and given talks on shipboard illumination. He represents the Bureau on committees of several national and international illumination societies. He is a member of the American Institute of Electrical Engineers and the Illuminating Engineering Society.

PRE-WORLD WAR I

THE FIRST known naval use of any light projector which could be called a searchlight was on board a Union warship blockading a southern port during the Civil War.

On a dark night a Confederate ship attempted to run the blockade in the belief that it was shielded from detection by the blackness of the night. Suspicion was aroused, however, on board the Union ship and the first searchlight used at sea was brought into use. From the Union ship there came a feeble, poorly reflected, widely divergent beam of light. It swung around, searching for the enemy and finally, faintly illuminated, he was discovered.

Crude as was this searchlight, it made a place in warfare for the projector which has since been developed to a remarkable degree of efficiency. This first searchlight was merely a "limelight" produced by playing an oxy-hydrogen flame on a candle of calcium oxide—this source of light being backed by a crude, spherical, polished, metal mirror which reflected only about 50 percent of the total light.

The next military use of searchlights was at the siege of Paris (1870-71) in which arc lights were used. Arc lights had previously found application in lighthouses and, because of the good results obtained there, were tried in searchlights.

In 1876, Colonel Mangin brought out his silvered glass reflector which gave excellent results and was the forerunner of modern searchlight mirrors.

In the winter of 1881-82, the Imperial Austrian Navy, by means of searchlights and machine guns defeated the Dalmatians in the Bay of Cattaro. The English fleet under Admiral Seymour, in 1882, by means of its searchlights, prevented the Egyptians from erecting water batteries at Alexandria on the night of July 11. The occupation of Sfax by the French and the landing of English troops at Ismailia were accomplished under the rays of searchlights. Searchlights were also used by the French Admiral Courbet in the Min River and in the Spanish engagements at Melilla.

By this time, the Naval searchlight was gradually being used for tactical purposes other than illumination and detection.

The English naval maneuvers in 1886 at Milford Haven pointed up the effectiveness of using squadron searchlights for blinding and confusing gunners on shore and disconcerting the fire from their batteries.

In the autumn of 1903, the U. S. Atlantic Fleet engaged in joint maneuvers with the Army at the eastern entrance of Long Island Sound. As the fleet neared Fort Michie, all of the fleet's searchlights were suddenly concentrated on the fort at a signal from the flagship, lighting up the fort so that the movement of individual men could be plainly seen, and the fleet passed into the sound without a shot being fired from the main batteries of the fort. On discussing the operation afterwards, the officers of Fort Michie stated that they were so blinded by the concentration of searchlights that they could see nothing in the foreground and could not have fired a ranged shot. They could not pick out individual searchlights but could see only a continuous glare.

In 1907, the importance of searchlights for naval purposes was already widely recognized. The uses of searchlights were, first of all, to detect the enemy's ship at the greatest possible distance and, in the second place, in the case of torpedo boat attack, to illuminate the surface of the water over a large area horizontally. It was only through the aid of searchlights that it was possible to repel torpedo boat attacks on dark nights. The blinding beam of light properly handled by defenders increased the difficulties of attack considerably and had a demoralizing effect on the attackers. The pupil of the eye, which was opened wide due to darkness, allowed such a quantity of light to fall on the retina when the searchlight beam was trained on the torpedo boat that the retina became over-excited and the attacker was incapable of seeing clearly for some time, even when the beam of light had been turned away. Thirdly, the searchlight was to serve as a signalling apparatus to transmit commands over great distances; and fourthly, to illuminate the immediate vicinity for landing parties or when passing through narrow waterways.

The desire of having, when required, either a concentrated beam or a dispersed one, led to the development of a double dispenser which was first used in the German Navy about this time. With this innovation it was made possible to produce at will either a concentrated beam of light or to pass to any degree of dispersion up to 45 degrees. The searchlight could then be used either for near or for distant illumination.

WORLD WAR I

During World War I destroyers were sent out from Scapa with orders to use their searchlights after dark in order to force submarines to keep under water and so exhaust their batteries. The search proved abortive, however, although it was continued for 24 hours. The use of searchlights for this purpose, begun on this occasion, became common later on.

Searchlights were an important factor during the Battle of Jutland in 1916. The following is a partial account of this battle by Admiral Viscount Jellicoe which deals with the part that searchlights played:

"When Captain Stirling had located the enemy's battle squadron he reported the fact by wireless but the signal was, unfortunately, not received by any ship, owing, presumably, to the strong interference caused by German wireless signalling at that time.

"At 5:56 p.m. Admiral Sir Cecil Burney reported strange vessels in sight bearing south-south-west and steering east, and at 6:00 p.m. he reported them as British battle cruisers three to four miles distant, the LION being the leading ship.

"This report was made by searchlight and consequently reached me after 6:00 p.m., but as showing the interval that elapses between the intention to make a signal and the actual receipt of it (even under condition where the urgency is apparent, no effort is spared to avoid delay and the signal staff is efficient), it is to be noted that whereas the report gave the bearing of our vessels as south-south-west, notes taken on board the COLUSSUS placed our battle cruisers one point on the starboard bow of that ship, that is, on a south-south-east bearing and distant two miles at 6:05 p.m."

"In the first place, such a course must have inevitably led to our battle fleet being the object of attack by a very large destroyer force throughout the night. No senior officer would willingly court such an attack, even if our battleships were equipped with the best searchlights and the best arrangements for the control of the searchlights and the gunfire at night."

"It was known to me that neither our searchlights nor their control arrangements were at this time of the best type. The fitting of director-firing gear for the guns of the secondary armament of our battleships (a very important factor for firing at night) had also just been begun. Without these adjuncts I knew well that the maximum effort of our fire at night could not be obtained and that we could place no dependence on beating off destroyer attacks by gunfire."

"The great efficiency of German searchlights at the time of the Jutland action and the greater number of torpedo tubes fitted in enemy ships, combined with his superiority in destroyers would, I knew, give the Germans the opportunity of scoring heavily at the commencement of such an action."

"There is no doubt at all that the German organization for night action was of a remarkably high standard. In the first place, the use of star shells, at that time unfamiliar to us, was of the greatest service to them in locating our destroyers without revealing their own positions; and secondly, their searchlights were not only very powerful (much more so than ours) but their method of controlling them and bringing guns and searchlights rapidly on to any vessel sighted was excellent. It also appeared that some system of director-firing was fitted to the guns of their secondary armament.

"At 11:00 p.m. the light cruiser ACTIVE, astern of the 2nd Battle Squadron, observed a ship coming up from astern, and shortly afterwards, saw searchlights switched on and a heavy fire opened against this vessel by a ship, or ships, on her starboard quarter. She appeared to be heavily hit and to sink. It is possible that this ship may have been the BLACK PRINCE which had apparently lost touch with our fleet during the day action."

WORLD WAR II

In the early part of World War II, searchlights were used much more often than during the latter part of the war when security was deemed more important. The following accounts of two engagements in this war show how effective searchlights were when used under certain conditions.

Battle of Matapan, March 1941 (Associated Press account)

A few seconds later one of our destroyers, the GREYHOUND, switched her searchlight on a third ship in the enemy's lines. We recognized her as one

of the 8-inch gun cruisers of the ZARA class. Subsequently, it was learned she was the FIUME.

The WARSPITE's 15-inch guns at once opened fire at very close range. The first broadside struck the cruiser with devastating effect. The whole ship, from foremost funnel to aft, burst into a mass of flames. The after turret was blown clear over the side. After one more broadside caused a fresh outburst of fire, the ship was hopelessly crippled.

We switched fire to the next cruiser in line, the ZARA. Again the first salvo produced an incredible effect. She burst into flames.

By now the entire battle fleet was in action. Destroyers sank the cruiser POLA. Our searchlight illuminated two enemy destroyers. These were ideally placed for torpedo attacks against us.

A 21-year old gunner on the FIUME, one of the few on duty when the smashing British attack was launched said:

"We were cruising along about 9:00 p.m. Friday night when suddenly in the distance, about three or four miles away, big searchlights flashed on us."

"For a few seconds we thought they were the lights of another Italian cruiser although we had sighted one British warship just before dark."

"A salvo of shells almost immediately followed the searchlight and simultaneously we were hit by a torpedo."

"Before we could bring a single gun into action the ship was listing too badly to fire."

Night Sea Battle off Guadalcanal, August 1942

"Our guns were blasting back but they ceased firing fearing they might hit one of our own ships. At the mouth of the channel one of our destroyers flashed its searchlights, picking up a Jap cruiser. The Jap cruiser in turn blinked its searchlights on the destroyer. It looked like a Hollywood premiere. The destroyer fired five torpedoes and one hit was certain."

With the advent of radar, the nighttime use of searchlights under battle conditions was virtually discontinued. The principal applications of searchlights were for daytime signalling and for search and rescue work at night although, during the last days of the Pacific campaign, some attempts were made to counter Kamikaze attacks by blinding the Japanese pilots with searchlight beams.

POSTWAR

In 1946, the U.S. Navy embarked on a program to reduce the weight and increase the signalling range of its searchlight. It appeared that the most significant gain would be made by using light sources of greater luminous efficiency which could be operated from an alternating current supply.

About this time some work had been done in Germany and in England on short-arc, high-intensity xenon and mercury arc lamps.

The U.S. Navy concentrated its initial effort on a 1 KW mercury arc lamp for its 12-inch signalling searchlight, the work horse of the fleet. Because of the scant background of knowledge and experience in the design of such lamps, many years were required to overcome the design and production difficulties. In 1951, the first prototype models of the 12-inch searchlight with the mercury arc lamp were evaluated by the fleet. Today this light and other signalling lights using similar type lamps are standard items in the Navy and are contributing to improving the effectiveness of the fleet.

The outstanding postwar improvement in searchlights has been the new light sources—the mercury arc and the xenon arc. Because of the constantly increasing amount of message traffic at sea, radio channels are becoming more and more congested and the need for greater reliance on methods of communication by visual means is becoming increasingly evident. Also, during periods of radio silence, long range communication may be carried out only by visual methods. In view of these increased needs, it appears that the signalling role of searchlights will become more significant in the future. One limitation of current designs is the mechanical shutter which is limited in speed and accuracy by the operator. If searchlights are to handle a heavy and sustained volume of message traffic, then some means of transmission other than by mechanical shutter appears necessary. One method of accomplishing this is by modulation of the new sources which will permit automatic transmission and reception at very high speeds and will greatly increase the efficiency of searchlight communication.

BIBLIOGRAPHY

1. Searchlights and Signal Lights—Navy Dept., Bureau of Steam Engineering, 1918.
2. Electric Ship Lighting—John W. Urquhart, 1902.
3. Searchlights—Their Theory, Construction and Applications—F. Nerz, 1907.
4. Proceedings of the U.S. Naval Institute—1894.
5. Searchlights Barrage in Night Landings—Capt. E. P. Jessop, USN (Ret)—U.S. Naval Institute Proceedings, June 1936.
6. The Grand Fleet, 1914-1916—Admiral Viscount Jellicoe, 1919.
7. The Battle of Jutland—Account by Admiral Viscount Jellicoe, Naval War College, 1920.
8. The *Washington Evening Star*—March 31, 1941.
9. The *Washington Evening Star*—April 3, 1941.
10. The *Washington Post*—October 28, 1942.

THE NOISE PROBLEM ON BOARD SHIP

ACKNOWLEDGEMENT

The May, 1956 JOURNAL *reprinted an article titled "A Review of Ship Vibration Problems" by W. Ker Wilson. The present article by the same author deals with an aspect of vibration which is assuming more and more importance in all classes of ships. This article was published in the January and February, 1956, editions of "The Marine Engineer and Naval Architect."*

ABOVE a frequency of about 1,800 cycles per minute vibration enters the audible range. These higher frequency vibrations are rarely dangerous from the point of view of the general hull structure, though they are able to cause considerable damage to machinery components, such as the teeth of a reduction gear assembly. The most serious consequence of excessive noise in passenger and crew accommodation is its effect upon comfort. In the engine room and on the bridge it can be a source of danger through its effect upon the efficiency of the supervisory staff. In extreme cases essential orders might be difficult to communicate and comprehend.

It has been said that noise is half the fun of Niagara. This remark illustrates one of the difficulties of establishing a truly comparative measure of the unpleasantness of audible vibration, namely the many factors which condition acceptance of a noise of given physical intensity as pleasant or unpleasant.

DECIBELS

The standard unit for measuring sound intensity is the decibel, which is not an absolute unit of intensity but merely expresses the ratio of a given intensity to an agreed reference value. But an intensity level of, say, 60 decibels does not produce twice the physical sensation of 30 decibles, nor do two intensities each of 30 decibels produce the same sensation as a single intensity of 60 decibels. This is partly because the smallest change which the ear can detect at a given intensity level is proportional to the prevailing intensity, i.e., the response of the ear follows the logarithmic law of growth; and partly because the response of the ear is a function of frequency. The decibel scale extends from the threshold of audibility (0 decibels) to the threshold of feeling or pain (120 decibels). At the higher intensity levels, from about 90 decibels upwards, the sensation produced is substantially independent of frequency, but at the lower levels frequency has a significant influence. At all intensity levels, the ear is most sensitive at frequencies in the neighborhood of 4,000 cycles per second (the higher notes of the violin or piccolo) probably because there are many natural sounds within this range to which the ear has become accommodated. It is therefore desirable, when investigating a complex noise pattern, to record the intensity levels and frequencies of the component sounds. This can be done by making a noise analysis in which, with the aid of acoustic filters, the intensity levels over narrow frequency bands are determined. By making such an analysis for an installation which has been judged to be acceptable by a number of observers, it might be possible to establish a reasonable acceptance standard. Furthermore, the analysis is a useful means for identifying the dominating components. This might lead to discovery of the principal contributory sources and so direct any necessary corrective action along the right path.

AIRCRAFT INVESTIGATIONS

Fig. 1 shows the recommendations published a number of years ago by the Royal Aircraft Establishment for the maximum intensity levels which should be permitted in aircraft.[1] These recommendations were based upon noise analysis over narrow frequency bands, and they represent the levels which it was considered could be achieved reasonably. They should therefore be regarded as minimum desirable values.

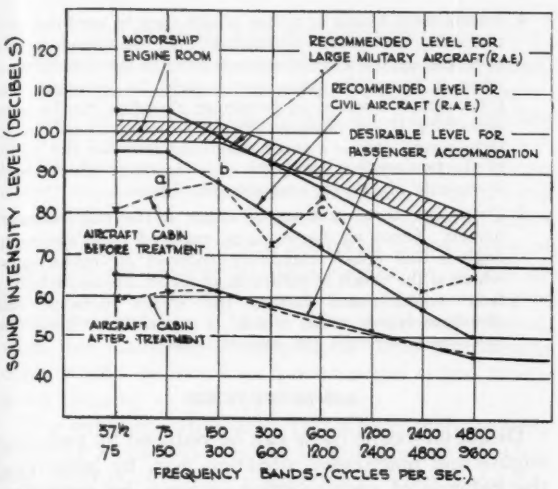


Figure 1. R.A.E. recommended maximum sound intensity levels for aircraft.

About twenty years ago an investigation was undertaken in the United States [2] to determine the best that could be done to quiet a twin-engined passenger airplane. The dotted lines in the diagram show the results of this investigation.

Before treatment the noise intensity in the higher frequency ranges was well above the R.A.F. recommended level for a civil aircraft, and it is interesting to note that the analysis shows clearly the effect of propeller blade frequency noise (point a); engine exhaust noise (point b); and the noise due to resonance of a skin panel adjacent to the propellers (point c). The sound-proofing treatment consisted of strengthening all panels where significant local vibration was detected; adding sound absorbing material to dampen residual panel vibration and absorb the higher frequency components of the noise; and adding cabin trimming materials to absorb the lower frequency components. The result was a very quiet airplane for an expenditure of weight for sound-proofing purposes of 280 lb. or 1.5 per cent of the gross weight of the aircraft. In 1940, the same reduction of from 20 to 30 decibels was being achieved with a weight penalty of slightly less than one per cent of the gross weight and the noise level was so good that some of the airline operators found it necessary to instruct their stewardesses to caution passengers on sleeper planes to speak quietly in order not to disturb those who were asleep!

It would therefore appear that the lower full-line plotted in the diagram represents a desirable maximum limit for the sound intensity in passenger accommodation. The suggested limits are 65 decibels at the low frequency end and 45 decibels at the high frequency end. In this connection an analysis of passenger complaints, undertaken by an American operator some years ago, showed that a noise level of about 75 decibels was generally accepted by ship and railway passengers, but when the level exceeded 75 decibels many complaints were received.

PHONS

When the overall intensity level of a complex sound pattern is under consideration, loudness is the characteristic which appears to be most closely related to the sensation produced. Measurement of loudness level is not an easy undertaking. Each measurement is really a laboratory experiment in itself. The procedure requires a comparison to be made between the noise and a standard reference sound which can be adjusted until the loudness of the two, as judged by an observer with normal hearing, appears to be the same. The standard reference sound is a pure tone of 1,000 cycles per second frequency, the intensity of which is increased until the loudness appears to be the same as the loudness of the noise under investigation. This comparison can be made, for example, by listening simultaneously to the standard tone with one ear and the noise with the other.

MEASURE OF LOUDNESS

The amount by which the intensity of the standard tone has to be increased to match the noise is measured in decibels above the threshold of audibility but, to avoid confusing measurements of intensity level with measurements of loudness level, the units are called Phons when they represent loudness. This definition indicates that if the noise whose loudness level is being measured is itself a pure tone of 1,000 cycles per second frequency then the loudness level expressed in phons is the same as the intensity level expressed in decibels. If, however, the noise is a pure tone of only 100 cycles per second frequency, the loudness level corresponding to an intensity level of 50 decibels would fall to 20 phons, due to the reduced sensitivity of the ear to the lower frequency sounds. Many attempts have been made to develop an instrument capable of giving a direct measurement of loudness level, but none has been found completely satisfactory over the whole range of frequencies. The method of comparison just described is probably the best available means of obtaining an approximate indication of overall loudness, though in the final assessment the ear itself must remain the real arbiter of nuisance values.

In Fig. 1, the overall loudness levels corresponding to the recommended intensity levels are 109 phons for large military aircraft, 98 phons for civil aircraft, and 62 phons for the desirable level in passenger accommodation.

It is of interest to note that if the intensity level in passenger accommodation remained constant at 65 decibels at all frequency bands, then the overall loudness level would increase to 72 phons. This indicates the importance of the higher frequency components in determining overall loudness level.

The normal range of the ear extends from the threshold of audibility (0 phons) to the threshold of feeling or pain (120 phons). At a loudness level of 60 phons conversation is possible in low tones over distances of 24 feet; at 70 phons conditions are still

comfortable, and conversation is possible in normal tones up to distances of 12 feet; at 80 phons conditions are fairly comfortable, and conversation is possible with slight effort; at 90 phons conditions are becoming uncomfortable, though conversation is still possible with some effort; at 100 phons conditions are definitely uncomfortable and conversation is difficult; at 110 phons conditions are becoming painful and conversation is impossible. In general a loudness level of 60 phons represents a good standard of comfort, though, as already stated, the ear is the final arbiter.

In marine applications, noise levels as high as 115 phons were reported a few years ago, from the engine-rooms of naval vessels fitted with geared steam turbine machinery. Improvements in the accuracy of gear tooth cutting and finishing have reduced this to about 102 phons, which represents a reduction of loudness level to about one-third of the original value. In the case of the less highly-rated gear assemblies of merchant vessels a value of 97 phons has been attained which is one-fifth of a loudness level of 115 phons.

In this connection Bunyan reports [3] a case where in some particularly noisy double-reduction main gearing assemblies of fabricated construction built during the war, a reduction of about 16 decibels in gear noise was obtained by lagging the gear case to the depth of the stiffening ribs with a heavily-loaded plastic compound retained by light sheet metal cover plates.

THE NOISE PROBLEM ON BOARD SHIPS

A typical distribution of sound intensity levels with frequency for the engine-rooms of vessels fitted with internal combustion engines is shown by the cross-hatched band in Fig. 1. The corresponding overall loudness level is from 103 to 110 phons, so that for comfort the loudness level in passenger and crew accommodations should be reduced to about 50 phons below the loudness level in the engine room. The first systematic studies of the noise problem on board ship were undertaken nearly twenty years ago.[4] Since then a considerable amount of valuable detail knowledge has accumulated, but the fundamental principles have remained substantially unchanged. These may be summarized as follows:

1. In dealing with the general problem of noise reduction in a particular application, such as on board ship, half measures are likely to produce very disappointing results, out of all proportion to the cost and trouble taken. *A large amount of noise can leak through an uncovered keyhole.*
2. This does not apply, of course, in circumstances where abnormal noise is due to some derangement of the machinery, such as running gear out of adjustment or bolted connections working loose. In these circumstances the excessive noise is a timely warning of impending trouble.
3. The higher frequency components of a complex noise pattern such as the high-pitched squeal of a noisy reduction gear or the scream from a high-speed fan, are more distressing than the lower frequency components, especially if they are in the region where the ear is most sensitive (about 4,000 cycles per second).
4. Intermittent noises or noises which vary in loudness periodically are more distressing than sustained noises even though the actual loudness level of the intermittent noise is lower. In passenger vessels, for example, the intermittent working of passenger elevators can be very disturbing if the noise penetrates to adjacent cabins. In twin-screw vessels a small difference between the r.p.m. of the two engines can produce a "beating" effect which is capable of causing considerable distress.
5. There are three sources of noise in the engine room, namely, direct air-borne noise radiated from the main engines and other machinery, indirect air-borne noise which is the result of reflection of direct air-borne noises from engine-room casings and other surfaces, and structure-borne noise which is transmitted from the machinery through the hull structure.

AIR-BORNE NOISE

Direct air-borne noise can be reduced by reducing engine and machinery vibration, e.g., by improving the balance of reciprocating engines, by providing uniform and smooth combustion in the cylinders of internal combustion engines, and by insisting upon the highest quality of workmanship in the manufacture of gearing assemblies. The weight of the *moving* parts of machinery should be kept as small as possible to minimize inertia forces which can be the cause of much clatter. On the other hand massiveness of the *stationary* parts is an advantage since it tends to reduce vibratory amplitudes. Experience shows, for example, that noise from an internal combustion engine increases as the specific weight of the engine is reduced. The stationary parts should also be made as rigid as possible to prevent excessive local vibration due to reasonance. Curved surfaces are better in this respect than large flat areas and unsymmetrically-disposed stiffening ribs are better than symmetrical patterns, points which are worth keeping in mind in connection with the design of large gearboxes and engine crankcases.

Direct air-borne noise can be reduced by fitting an acoustic hood over the machine. This can consist of a sheet metal cover lined with sound-absorbing material fitting closely over the machine and easily removable to give access for maintenance. There should be no direct metallic contact between the hood and the engine or ship's structure, to prevent the hood itself from becoming a source of noise radiation through transmission of structure-borne vibration. In this connection an interesting example was reported a number of years ago where the noise from a reduction gear assembly was substantially reduced by removing the gearcase cover bolts and supporting the cover upon a number of matchsticks inserted in the joint. This was an example where the noise due to gear tooth engagements was being transmitted to the cover through the gearcase structure.

Valve gear noises in internal combustion engines is less when the mechanism is totally enclosed, and chain drives are quieter than gears.

EXHAUST AND INTAKE NOISE

Other important sources of direct air-borne noises in the engine-rooms of ships fitted with internal combustion engines are the engine air intake and the exhaust systems.

The minimum frequency which the normal ear can detect as a continuous sound is about 30 cycles per second. Thus the suction noise from an engine-driven scavenge pump, which occurs at a frequency of one cycle per revolution of the engine, would not be heard as a continuous sound in engines running at less than 1,800 r.p.m. In the case of a rotary blower, with, for example, a three-lobed rotor the suction noise would be heard as a continuous sound in engines running at 600 r.p.m. or more.

The exhaust impulses of two-stroke cycle engines occur at a frequency equal to the number of cylinders multiplied by the engine r.p.m. Thus, for a six-cylinder two-stroke cycle engine the exhaust note would not be heard as a continuous sound in engines running at less than 300 r.p.m.

In four-stroke cycle engines the exhaust impulse frequency is one-half the number of cylinders multiplied by engine r.p.m., so that the exhaust note of a six-cylinder four-stroke cycle engine would not be heard as a continuous sound in engines running at less than 600 r.p.m.

However, the intermittent nature of both suction and exhaust produces shock excitations which are heard as a low frequency throbbing noise which can be very distressing. In many cases the exhaust throb appears to have a frequency equal to the r.p.m. of the engine, because the impulse from the cylinder nearest to the exhaust outlet has a masking effect upon impulses from the other cylinders.

An analysis of intake and exhaust noise shows that it contains both high and low frequency components, the latter generally predominating.

The higher frequency components can be damped by lining the inlet and exhaust manifolds with sound absorbing material, a method which can be quite effective at frequencies above about 250 cycles per second.

Other methods are required, however, for dealing with the lower frequency components, particularly noticeable in the larger engines.

Sheer weight of material in the walls of exhaust manifolds is beneficial and for this reason cast iron manifolds with cooling water jackets are quieter than manifolds made from sheet steel, even when the latter are well lagged for heat insulating purposes. Where weight is an important consideration, however, the lower frequency components of the exhaust and intake noise can be reduced by connecting the interior of the manifold to one or more annular channels surrounding the manifold and forming enclosed air cavities. These act as resonators and are particularly effective at their resonant frequency, though they will also deal to some extent with noise components of higher frequency. If the manifold is connected to the atmosphere this is virtually the same as connecting it to an infinite cavity for which the resonant frequency is zero. This device has therefore proved effective for reducing the very low frequency noise from air intakes. Resonating cavities are principally of value for dealing with noise components in the frequency range 30 to 300 cycles per second. Above 300 cycles per second absorbent materials are better.

Fig. 2 shows three designs of exhaust silencer based upon noise investigations carried out on motor-ships in German shipyards. They are of the resonating cavity type, either with or without linings of sound-absorbing materials. Silencers of this type were used on the German pocket battleships. Fig. 3 shows three designs of air-intake silencer. The arrangement at (a) consists of a simple tube lined with sound-absorbing material and connected to the atmosphere by three rows of holes. It is therefore specially suitable for dealing with the very low frequency noise from reciprocating scavenge pumps.

Design (b) shows an ingenious arrangement of venturi nozzles. If the air velocity in the throats of the nozzles is equal to the velocity of sound in air, no sound can be reflected from inside the silencer. In practice, to avoid excessive friction loads, the throat velocity is usually well below the velocity of sound. Nevertheless the device remains quite effective for dealing with low and medium frequencies.

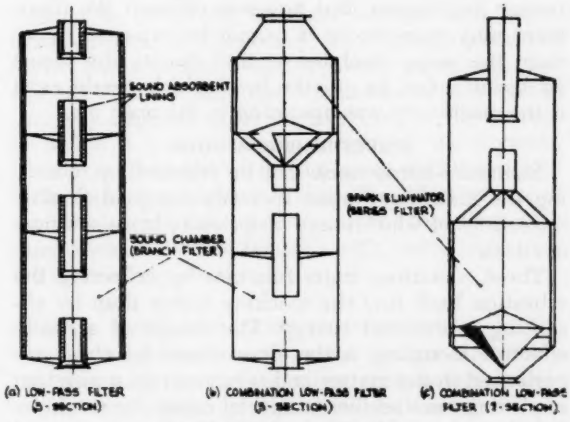

SOUND ABSORBENT LINING
SOUND CHAMBER (BRANCH FILTER)
SPARK ELIMINATOR (SERIES FILTER)
(a) LOW-PASS FILTER (5-SECTION)
(b) COMBINATION LOW-PASS FILTER (5-SECTION)
(c) COMBINATION LOW-PASS FILTER (1-SECTION)

Figure 2. Exhaust silencers.

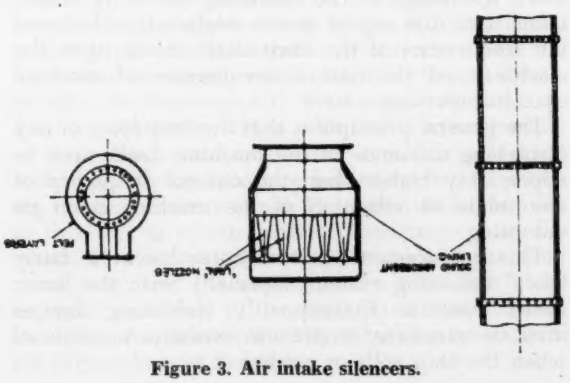


Figure 3. Air intake silencers.

TURBOCHARGER WHINE

Design (c) uses sound absorbing linings which are specially suitable for dealing with the high-pitched whine from turbo-superchargers, which has a dominant frequency equal to the number of vanes on the supercharger impeller multiplied by the r.p.m. of the impeller. It is useful to note that an appreciable reduction of loudness can be obtained by selecting the number of impeller vanes so that the dominant frequency is outside the range in which the ear is most sensitive (3,000 to 5,000 cycles per second). This implies that it is desirable to select a number of vanes outside the values n=180,000/N to 300,000/N, where n is the number of vanes and N is the r.p.m. of the impeller.

Indirect air-borne noise in the engine room arises from reflection of direct airborne noise from the engine room casings and bulkheads. The amount reflected depends upon the rigidity of the reflecting surface and it is perhaps not generally appreciated that the efficiency of reflection from a solid surface is very high, superior to that of light from the best mirror surface.

The amount of reflected noise can be reduced by lining the reflecting surfaces with sound absorbing material. Some highly efficient proprietary materials are now available for this purpose and the manufacturers have accumulated a considerable amount of information on the treatment best suited to a particular application. But however efficient the treatment may prove to be it cannot be expected to reduce the noise level below that due to the direct air-borne noise, i.e., to the level which would exist if the machinery was operating in the open air.

STRUCTURE-BORNE NOISE

Structure-borne noise can be reduced by mounting the machinery upon specially-designed flexible mountings of which many proprietary types are now available.

These mounting units function by reflecting the vibration back into the machine rather than by absorbing vibrational energy. The design of a really effective mounting is therefore a task for those experienced in the matter and is not merely a question of placing some resilient material under the machine. Satisfactory results are only likely to be obtained when the design of the mounting system is undertaken with due regard to the relationships between the frequencies of the excitations acting upon the machine and the natural frequencies of machine upon its mounting system.

The general principle is that the frequency of any disturbing influence in the machine itself must be appreciably higher than the natural frequency of any mode of vibration of the machine upon its mounting.

This requirement generally results in a fairly "soft" mounting system, especially with the lower speed machine. Consequently stabilizing devices must be introduced to prevent excessive movements when the ship rolls or pitches.

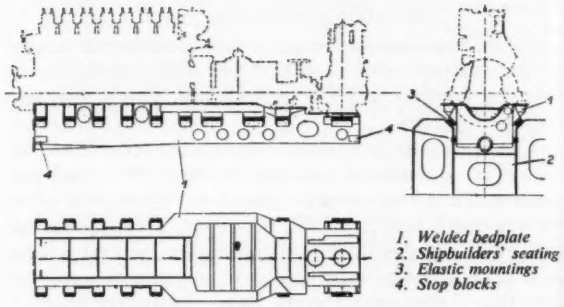


Figure 4. One method of carrying auxiliary generator sets on resilient mountings.

The disappointing results sometimes obtained from flexibly-mounted machines are generally due to leakage at places, such as inlet, exhaust oil and water pipe connections, where isolation has not been provided.

It is therefore important to make sure that all service piping is flexibly connected, not only at the machine itself *but at all points where it is supported from the ship's structure.* Also, to avoid excitation of local vibration, the support points should be at a rigid part of the structure.

Structure-borne noise can augment the overall noise level in the engine room by exciting vibration at local areas of the hull structure, especially if local resonance occurs. This is especially likely to occur in engine-room casings, bulkheads and floors. Sound-absorbing treatment applied to the free surfaces will not have much effect in reducing this type of vibration, which will usually require the introduction of some local stiffening.

LOUDNESS LEVEL IN ENGINE ROOM

So far as noise within the engine room is concerned the principal measures that can be applied are reduction of vibration of the machinery itself to the lowest possible level; reduction of direct air-borne noise by means of acoustical hoods; attenuation of exhaust and intake noise by acoustical treatment of exhaust and intake manifolds; reduction of indirect air-borne noise by application of noise absorbing materials to engine-room casings, bulkheads and other free surfaces; and reduction of structure-borne noise by flexibly mounting the machinery and its service connections, and by stiffening those areas of the ship's structure where local vibration is found to be excessive.

When all these measures have been applied it should be possible to reduce the overall loudness level in the engine-room of a ship fitted with internal combustion engines from the normal level of about 100/110 phons to about 95 phons.

For real comfort in the engine room, however, it would be necessary to reduce the overall loudness level to at least 75 phons, a target which is not very likely to be attained for the engine room as a whole.

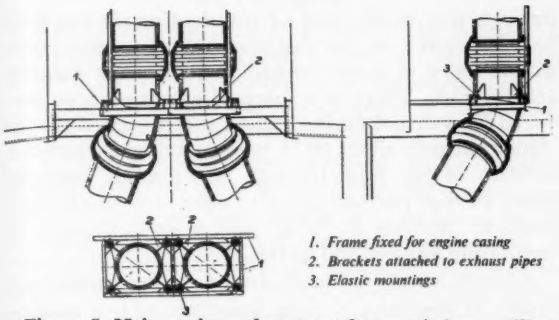


Figure 5. Main engine exhaust uptakes carried on resilient supports.

There would not be a great deal of difficulty, however, in attaining a level of 75 phons in a specially-constructed control room, resiliently supported from the ship's structure and provided with strong sound-insulated walls containing double windows of safety glass to provide a clear view.

This type of control room is now a standard feature of aero-engine test-beds where it provides a quiet region directly adjacent to the noisiest turbo-jet engine.

NOISE TRANSMITTED TO ACCOMMODATIONS

The noise transmitted from the engine-room to the accommodations is also both air and structure-borne. Airborne noise originates from the direct air-borne noise in the engine room impinging upon the engine-room casings and setting them vibrating. Some of the incident noise is transmitted through the ship's structure to the accommodations where it is radiated from the ship's plating or the cabin structure, particularly if there are conditions of local resonance. Vibration of the engine-room casings also radiates noise into the adjacent alley-ways along which it is readily conducted by reflection from one boundary surface to another, finding its way into accommodations and public rooms through openings and producing local vibration through the effects of resonance.

Reduction of direct air-borne noise in the engine room itself, by reducing the noise emitted by the machinery, is also beneficial in reducing the air-borne noise transmitted to the accommodations. It should be noted, however, that the application of sound-absorbing materials to the interior sides of engine casings and other free surfaces in the engine room, while beneficial from the point of view of the loudness level within the engine room, has little effect upon the noise transmitted from the engine room to the accommodations. The materials used for heat insulating purposes are probably just as effective in this respect.

Structure-borne noise is transmitted through machinery seatings and pipe connections directly into the hull structure through which it may be transmitted to remote parts of the ship, where it may be re-radiated as air-borne noise, especially at places where there is local resonance of the ship's structure.

FLEXIBLE PIPEWORK

Transmission of noise through the structure can be reduced by introducing structural discontinuities which reflect a considerable proportion of the noise which would otherwise be transmitted. These are best placed as close to the source as possible. Examples are well-designed flexible mountings for auxiliary generators and other machinery, flexible supports for inlet and exhaust and service piping,[5] and pads of resilient material at the foot and head of engine-room pillars.

Transmission of noise from one side of a partition to the other is largely dependent upon the weight and rigidity of the partition, i.e., the heavier the partition the less the amount of noise transmitted. Where lightness is an important consideration a composite construction can be used to increase the effectiveness of the insulation. For example, noise transmission through engine-room casings can be reduced appreciably by applying solid plywood panelling mounted upon wooden battens to act as stiffeners and provide an air space between the panelling and the casing. Direct contact between the panelling and the casing should be prevented by setting the supporting battens in rubber or other resilient material and the efficiency of the insulation can be improved by providing a lining of sound absorbing material, such as felted mineral wool in the air-space. The damping effect of the absorbent lining can be increased by attaching its inner surface to a heavy material such as fibreboard. The lining should not make contact with the plywood panelling. The amount of noise reduction which can be achieved in this way depends upon the liveliness of the surface to which it is applied, the weight of the plywood panelling, the width of the air-space, and the weight of the sound absorbing lining. The effectiveness of the treatment also depends upon the frequency ranges which have to be dealt with, being higher for the higher frequencies.

NOISE ANALYSIS RECOMMENDED

From considerations of weight and general economy, therefore, it is always desirable, before undertaking any scheme of sound insulation, to make a noise analysis to determine the frequencies of the predominant components. This preliminary work should also include a survey of the ship's structure to determine which are the lively surfaces where the insulating treatment would be likely to provide a worthwhile improvement. Such a survey may, indeed, reveal places where vibration is excessive due to some local resonance effect in which case local stiffening may be necessary before any sound-proofing treatment is attempted. The information obtained in this way is of the greatest importance from the points of view of both effectiveness and economy since it enables the quality of the treatment to be adjusted according to the requirements of different localities. An industrial stethoscope is a very useful aid in carrying out this task.

CABIN TREATMENT

Where the loudness level inside a cabin is excessive due to radiation of structure-borne noise from the hull surface the same type of sound insulating treatment should be applied, namely, rigid panelling supported upon wooden battens set in rubber or other resilient material, with a layer of sound-absorbing material inserted in the air-space between the hull surface and the panelling. Mere application of a layer of sound-absorbing material to the hull surface will not be effective in reducing the loudness level due to structure-borne noise, though it will reduce the loudness level due to any noise originating within the cabin itself, e.g., the noise of typewriters and printing machinery in offices, noise in fan rooms, and in reading and dining rooms.

The same remarks apply to cabin partitions, ceilings and floors.

In all these cases double-walled partitions with a layer of sound absorbing material between are more effective than a single solid partition of the same weight.

In all cases the panelling should be isolated from the ship's structure by setting the mounting battens in rubber or other very resilient material so that a series of floating walls, ceiling and flooring is provided, forming what is virtually a room within a room. It is also important to avoid direct contact between the edges of the various panels. These should be left free and the joints made by rubber inserts, remembering that rubber is only fully effective as an insulating material when there is freedom for elastic displacement to occur. For this reason rubber loaded in shear is much more effective than rubber loaded in compression.

In some cases a floating partition might vibrate quite appreciably in response to a low frequency excitation, below the audible range.

Though this might not be serious so far as the bulkheads and overheads are concerned, it might be disagreeable if it is picked up by the feet from a deck. It can be reduced by increasing the weight of the floating deck, e.g., by adding a covering of heavy material such as rubber carpeting, or by increasing its stiffness. It is desirable to arrange the battens, used for spacing panelling from the ship's structure, in an unsymmetrical pattern to avoid coincidence between a batten and a nodal line corresponding to a mode of natural vibration of the panel surface.

Sleeping berths must be effectively isolated from structure-born vibration. Cot beds, standing clear of the walls, and supported upon vibration isolating mountings are usually better in this respect than bunks supported cantilever fashion from the ship's side.

VENTILATING SYSTEMS

Another potentially serious source of noise transmission is the ventilating system. Noise originated by fan blades, motor hum, bearing impacts, and the rush of air through ducting, can be transmitted with high efficiency through the air stream itself and through the metal work of the ducting. It can also be transmitted to the ship's structure at any point where there is direct contact between the ducting and the hull. There is a sharp rise in the noise produced by ventilating fans when the tip speed of the blades exceeds about 60 ft. per second and when the velocity of air flow through the ducting exceeds about 20 feet per second. The noise due to air flow tends to increase at bends and at outlets such as ventilating louvres and grilles.

A fundamental requirement for a quiet ventilating system is therefore to keep velocities as low as economic and space considerations permit, and it is desirable to incorporate means for regulating the air supply according to demand. If this is not done there will be a tendency for the noise at individual cabin ventilators to increase substantially when ventilation is cut off at other points in the system.

The measures which can be taken to reduce the transmission and emission of noise in ventilating systems are relatively simple. The internal surface of the ducting should be lined with sound-absorbing material such as mineral wool, held in place by perforated sheet metal. The effectiveness of the treatment depends upon the ratio of the perimeter of the duct to its area, hence a narrow rectangular section is better than a square section. Where space considerations require a section which is substantially square, the channel can be divided into a number of narrow rectangular sections by means of splitters lined with sound absorbing material. Where sharp bends are unavoidable the formation of noise-making eddies can be prevented by using guide vanes to direct the air smoothly round the bend. The ducting should be isolated from the fan and from the ship's structure by flexible connections. Exit louvres and grills should be designed to minimize the formation of noise-making eddies.

As with other aspects of the noise problem, the precise treatment depends upon the nature of the particular noise pattern. Thus, thicker absorbent linings are required when the noise is predominantly low frequency.

PROPELLER-EXCITED NOISE

In general the propeller does not appear to be as important a source of noise as the main machinery of a vessel fitted with internal combustion engines. The minimum r.p.m. at which the fundamental note of a propeller, i.e., number of blades multiplied by propeller r.p.m., becomes audible are about 600, 450, 360 and 300 r.p.m. for 3-, 4-, 5- and 6-bladed propellers respectively. From this point of view therefore the propeller problem is one of mechanical vibration rather than noise.

There is, however, the possibility of audible vibration being excited at propeller blade frequencies through impacts and this no doubt accounts for the throbbing noise from the propeller which is sometimes heard in the after part.

SINGING PROPELLERS

The most difficult and elusive noise problem so far posed by the propeller was the phenomenon known as "singing," which first appeared about 20 years ago, with the general adoption of high-tensile alloys in propeller manufacture.

Though the phenomenon never really grew to epidemic proportions it was sufficiently serious to initiate research work which has continued to the present day.[7] This is not surprising when it is remembered that a "singing" propeller can produce a phenominal amount of noise, mostly in the middle range of the audio-frequency scale, which can generally be heard throughout the vessel. Even today the mechanism of "singing" is not fully understood, though there appears to be no doubt that it is due to excitation of a complex mode of vibration of the blade regarded as a vibrating diaphragm. Various theories have been put forward to explain the mechanism of excitation, including onset of cavitation which subjects the blade to a series of impacts due to the formation and collapse of cavities in the water, formation of trailing edge and tip vortices, shock excitation when the blades pass through a varying wake, and self-excited vibration due to interaction of torsional and flexural modes of vibration. It is also likely that the phenomenon is specially liable to occur when the mode of vibration is a reactionless mode for the propeller as a whole so that there is little or no motion at the boss and therefore damping is confined to the blades themselves. The magnitude of the impulses acting on the blade as it passes through a varying wake can be reduced by applying the same measures as are necessary to reduce mechanical vibration, namely, adequate aperture clearances in single-screw vessels and adequate blade to hull clearance in multi-screw installations; combined with an afterbody design which provides the smoothest possible flow of water through the propellers. In addition, the design-characteristics of the propeller blade itself can be adjusted to some extent to match the wake conditions; to delay the onset of cavitation, and by controlling the thickness of the blade edges and the plan form of the blade; to reduce the severity of vortex formation and increase hydrodynamic damping. During recent years a great deal has been learned about the vibration characteristics of propeller blades.[8] This knowledge can be applied to assist in tuning a given design so that modes of vibration favorable for "singing" can be removed from the operating speed range. In this connection, it is possible that the introduction of some dissimilarity between the vibration patterns of the several blades might be beneficial in preventing reactionless vibration of the propeller as a whole.

The result of all this effort has been to make "singing" of propellers a rare occurrence, and from accumulated experience designs can now be prepared with reasonable assurance that this trouble will not occur.

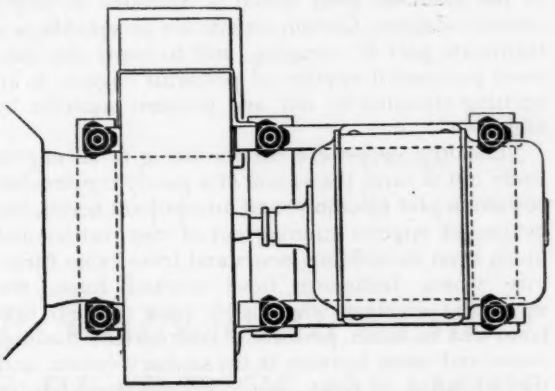


Figure 6. Ventilating fan unit mounted on six standard Silentbloc units.

MISCELLANEOUS

A great deal of engine-room noise can be air-borne to the deck spaces and public rooms on the upper decks from open, or even loosely-fitting engine-room skylights and doors. Other sources of deck noise are the whine from ventilator discharge ducts which have not received proper acoustical treatment and the throbbing from engine exhaust outlets where the silencing arrangements are not sufficiently effective. Where the engine is not unduly sensitive to back pressure a comparatively simple method of reducing exhaust outlet noise is to throttle the outlet by means of a discharge nozzle which changes the pulsating flow into a steady stream of gas.

Although the preceding discussion shows that the noise problem on board ship is exceedingly complex, the underlying principles of the subject are now sufficiently understood to enable valuable precautionary measures to be taken in the design stage of a new ship. Furthermore a considerable fund of practical knowledge is available from experience in the building and aeronautical industries as well as in the marine engineering industry itself, and from the results of the work of research establishments. Most of the measures which reduce the severity of mechanical vibration are also beneficial from the point of view of noise abatement. Indeed, there is no strict boundary between the two, and it may well be that progress will continue to depend to a large extent upon finding economical solutions to the vibration problems which each development inevitably brings in its train.

With regard to the present status of the noise problems on board ship, it is probably not too much to say that knowledge of causes and treatments is now sufficient to provide loudness levels not exceeding about 65 phons in the accommodations and 95 phons in the engine room, the matter being one of business economics rather than engineering skill—the Specification rather than the Text-book.

Complete silence, the eerie lifeless atmosphere of a sound-proof room, is as much to be avoided as excessive loudness and there is probably a lower limit

to the loudness level which is desirable in ship's accommodations. Certain sounds are acceptable as a legitimate part of voyaging, and to many the subdued purposeful rhythm of powerful engines is an exciting stimulus by day and pleasant soporific by night.

Unhealthy noises are the throbbing of an engine badly out of tune, the squeal of a poorly constructed reduction gear assembly; and intermittent noises, the beating of engines running out of step, rattles and thuds from loose-fitting doors and from cabin furniture where fastenings have worked loose, the squeaking of pipes where they pass through raw holes cut in cabin partitions, intermittent flushing noises and water hammer in the sanitary system, and the operation of noisy passenger elevators late at night adjacent to accommodations.

These disturbances are capable of causing distress out of all proportion to the loudness levels involved, and if the passenger finds cause to complain about any of them the man to send for is the engineer or the naval architect, certainly not the acoustic specialist.

REFERENCES

1. R.A.E. Report S.M.E. 3359, Leach and Cardwell.
2. "Comfort in Air Travel," S. J. Zand, Jl.R.Ae.S., July, 1936.
3. "Practical Approach to Some Vibration and Machinery problems in Ships," T. W. Bunyan, Trans. I.Mar.E., Vol. 67, 1955, p. 99.
4. "Noise Reduction in Ships," R. S. Robinson, Trans. I.Mar.E., Vol. 50, 1938, p. 259.
5. See: *Marine Engineer and Naval Architect*, Jan., 1955, p. 7.
6. "The Problem of the Singing Propeller," Kerr, Shannon and Arnold, Proc. I. Mech.E., Vol. 144, 1940, p. 54.
7. "Propeller Blade Vibration Tests," L. C. Burrill, Trans. N.E.C. Instn., Vol. 65, 1949, pages 273 and D.51.
8. "Underwater Propeller Vibration Tests," L. C. Burrill, Trans. N.E.C. Instn., Vol. 65, 1949, pages 301 and D.51.

J. J. FRANCIS
LT. R. E. KOSIBA, U.S.N.
R. A. WOOLLACOTT

THE ALIGNMENT OF MAIN PROPULSION SHAFT BEARINGS IN SHIPS

MR. J. J. FRANCIS

is a supervising naval architect in charge of the Scientific and Test Branch, Design Division, Boston Naval Shipyard. He holds both a Bachelor and a Master of Science Degree conferred by Massachusetts Institute of Technology in 1938 and 1939 respectively. He is a member of the Society of Naval Architects and Marine Engineers, and has co-authored a professional paper on Vibration Engineering published in the 1953 transactions of that Society. In all, Mr. Francis has eighteen years of government service, sixteen of which have been with the Navy Department.

LT. R. E. KOSIBA, USN

at the writing of this paper, was Assistant Design Superintendent at the Boston Naval Shipyard. He is a graduate of the U. S. Naval Academy. He attended post-graduate school at the Massachusetts Institute of Technology, and received his Naval Engineer's Degree from that institution in 1953. He is a member of the Society of Naval Architects and Marine Engineers and the Society of Sigma Xi. The Bureau of Ships, Propeller and Shafting Branch, Navy Department, Washington, D. C., is his present duty station.

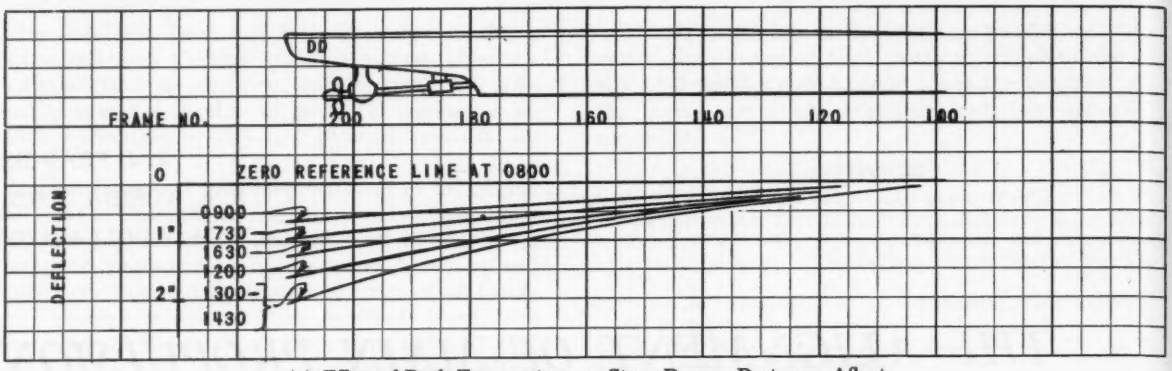
MR. R. A. WOOLLACOTT

is a supervising naval architect in charge of the Scientific Section of the Boston Naval Shipyard. He received a Bachelor of Science Degree in Engineering from Northeastern University in 1941, and subsequently studied Naval Architecture and Marine Engineering at Massachusetts Institute of Technology. He has worked as a Naval Architect at the Boston Naval Shipyard for thirteen years.

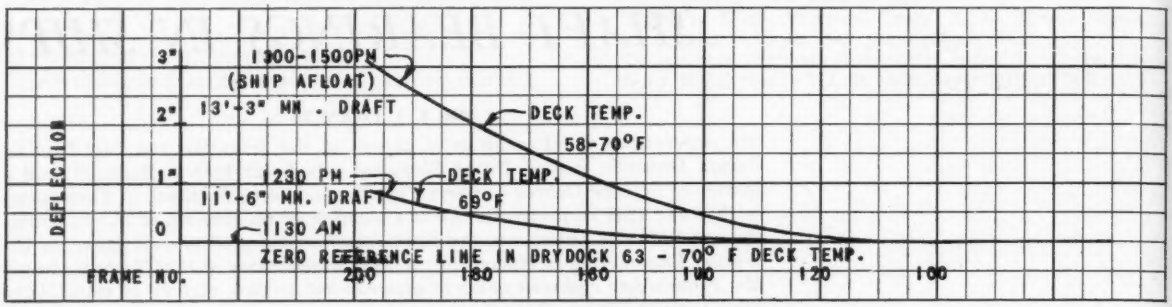
THE PROCEDURES employed at the Boston Naval Shipyard for accomplishing shafting alignment repairs are the results of years of experience in this field. Accomplishing such repairs efficiently has required close integration of the skills of mechanics and the theory of engineers. Though this paper does not attempt to prescribe such information as bearing wear or positioning tolerances, it does describe an approach by which for a particular ship the significance of these controlling parameters can be determined. The background and factors involved in shafting alignment are discussed together with comments on the measurement techniques in use, but the significant contribution of the Shipyard's approach is the manner in which the continuous beam theory has been applied to the shafting system as a whole (including attached populsion components such as bull gears and main motors). Case histories containing the results of the application of this approach to ships overhauled at the shipyard are presented as examples of the relative importance of various factors causing differential bearing movement.

Factors which indicate the need for thoughtful consideration of alignment practices include:

a. Cost (time and money) of alignment repairs, attributable to designed bearing arrangements and unrealistic tolerances for bearing positions.
b. Deleterious effects observed to attached propulsion components.
c. Unreal bearing reactions used in present bearing design and shaft stress analysis.



(a) Effect of Deck Temperature on Stern Droop—Destroyer Afloat



(b) Effect of Undocking on Stern Droop—Destroyer

Figure 1. Ship Deflection Due to Heating and Undocking

SHAFTING ALIGNMENT DEFINED

Shafting alignment consists of positioning bearings to support the shaft system to allow it to propel the ship in conjunction with the propulsion machinery. It is considered in four parts (1) static support of the weight of the shaft system (2) prevention of buckling of the shaft column (3) prevention of undesirable vibration and (4) the maintaining of working stresses within design criteria. This paper does not consider composite stress in the shafting.

BACKGROUND

Factors Influencing Shafting Alignment

Appreciating the environment of the shafting system is a pre-requisite to the understanding of shafting alignment. To this end, studies were made by the shipyard to establish the response of a ship's hull in way of shafting to such influences as temperature, loading, speed, etc.

The factors considered to influence shafting alignment are the position of the bearings, the distribution of the supported mass, and the forces imposed by dynamic action of the propeller, hull and attached machinery. The various effects which establish the position of the bearings are separated for this discussion into movements of the hull and movements relative to the hull.

Hull Movements: It is well known that temperature variations cause the hull girder to deflect both vertically and horizontally. Measurements (Figure 1a) revealed hull diflections between midships and the main strut of 1½ inches for a destroyer type ship (ship afloat).

Undocking changes also produce deflections of the hull girder. Measurements (Figure 1b) taken while undocking a destroyer hull revealed that the main strut section moved vertically relative to midships by 3 inches. (Temperature and loading were essentially constant). Changes in loading will produce similar hull deflections.

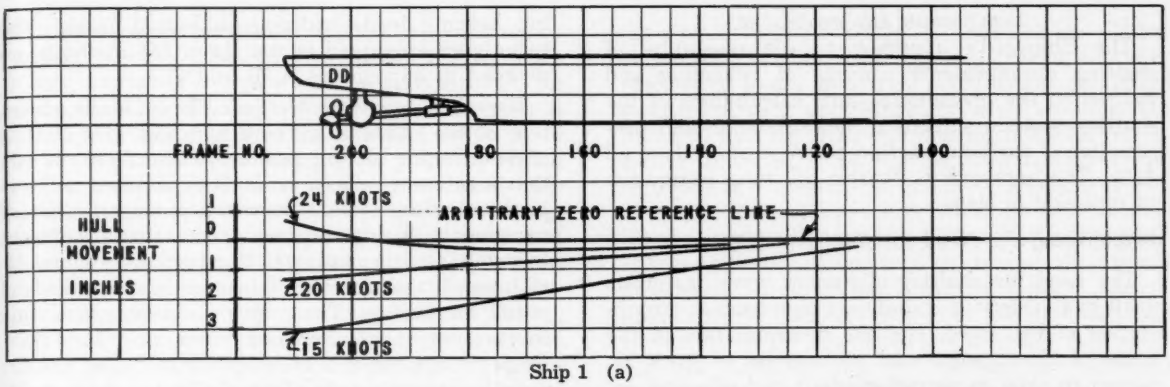
Vertical displacements (Figures 2a and 2b) were measured as a result of speed change at sea.

In appreciating the effects described above, it is apparent that in repair alignment needless time and money may be expended repositioning bearings to place them on a straight line, if the effects mentioned above will make any straight line relative to the hull conform to the deflection curve of the hull. It is considered that the hull deflects as a fair curve.

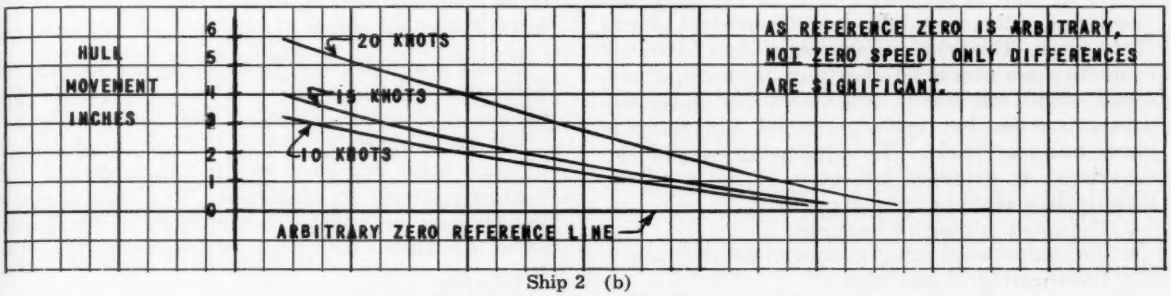
Bearing Movements Relative to Hull: In the operation of a ship, bearing wear produces an effective movement of the shaft support point relative to the hull. The magnitude of wear experienced varies in individual cases, but acceptable limits should be established by considering the requirements of the bearing manufacturer and the effect on the shafting system and associated equipment.

Inaccuracies in installing and moving bearings constitute an effective bearing movement. A design which requires positioning accuracy not attainable is certain to result in unsatisfactory operation and expensive maintenance.

Temperature also causes movements of bearings relative to the hull. The obvious offender in this



Ship 1 (a)



Ship 2 (b)

Figure 2. Movement of Stern with Changes of Speed

category is the reduction gear casing which has a lubricating oil sump built integral with the base. Measurements (Figure 3) of one casing showed a vertical lift of 0.025 inches of the bull gear bearings relative to the hull girder.

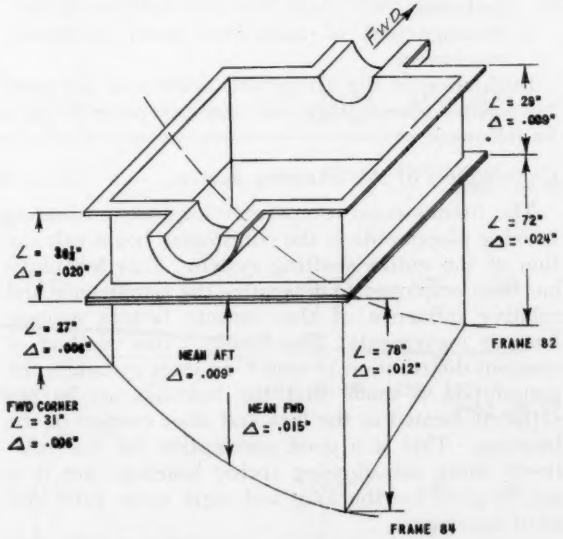
Other movements encountered result from repair to bearing housings, bearings, or associated structure. Inadvertent but significant movement is possible when a stern tube bearing housing is restored to dimension after excessive erosion or corrosion. Similarly, renewing one worn bearing may result in disastrously disturbing a system which has accommodated itself to misalignment by helpful wear down.

In appreciating the effects described above it is apparent that the bearing movements relative to the hull are characterized by a randomness. Unlike the fair curve deflection of the hull, a plot of the displacement of each bearing relative to the hull would be erratic.

Distribution of Mass: The distribution of the mass of the system and the longitudinal positions of the bearings are established early in design by the arrangement chosen for the propeller, shafting and attached machinery components. This is usually based on vibration and functional considerations. A lack of consideration of alignment requirements at this stage could result in a system so sensitive that continuous troubles would be experienced in normal operation. The examples further in this paper are illustrative.

Dynamic Loads: Generally, dynamic loads caused by ship motion in a sea way or by non-concentricity of propeller thrust are not considered.

Analysis of vibration complaints has revealed shaft whip because bearing spans were too large in the original design or because misalignment had rendered an installed bearing ineffective.



∠ = Length
△ = Displacement measured with Indicator

Total expansion to bearing C.L. Forward—.024" Aft—.029"
Note: Oil circulated for a period of 9 hours and temperature increased from 65° to 135° F.

Figure 3. Expansion of Port Reduction Gear Base due to Heating—Ship B.

EVALUATION AND PROCEDURE

The Shipyard's approach to the resolution of shafting derangements consists of collecting and evaluating the operational data, calculations of the shafting system, shipboard measurements and prescription of corrective action based on analysis of all data. The approach is flexible allowing reiteration or omission of steps.

Establishing the Need for Shafting Alignment

The need for shafting alignment work manifests itself in a variety of operational phenomena. Appreciation of this need, and the determination of the scope of investigation and the amount of work required in time to permit efficient and effective repair is one of the most difficult and important phases of the work. The principal sources of data for initial evaluation are contained in reports from ship's personnel, ship's machinery logs, inspections and measurements made by repair personnel and observations on sea trials. The following would be considered as evidence indicative of the need for investigating shaft bearing alignment:

a. Repeated wiping of bearings.
b. Excessive wear rate, or uneven bearing wear.
c. Overheating of bearings.
d. Damaged shaft, bearing or hull in way of shaft.
e. Unusual noises in the shaft system such as thumping or squealing.
f. Poor seating of shafting in bearings as indicated by clearance readings.
g. Leaking at stern tube stuffing box, or excessive working of stuffing boxes.
h. Whipping of shaft or vibration of shaft bearing pedestal.
i. Misalignment of associated main propulsion machinery.

Evaluation of the above will determine the need for further investigation and indicate procedures to be followed.

Calculations of the Shafting System

The fundamental comparator used for evaluating bearing placements is the continuous beam calculation of the entire shafting system. This technique has been employed to determine the significance and relative influence of the various factors causing bearing movements. The Hardy Cross method of moment distribution is used.* In most instances the assumption is made that the bearings act as pin supports located at the fore and after centers of the bearings. This is a good assumption for the relatively short, self-aligning spring bearings, but it is not as good for the long and rigid stern tube and strut bearings.

Shipboard Measurements

Three methods are employed for establishing bearing positions or reactions: (1) combination boresight and wire method, (2) the method of "drops and openings" and (3) a method of measuring bearing loads utilizing calibrated jacks. The techniques employed in the three (3) methods are detailed in appendixes A, B and C.

Boresight and Wire Method: The inherent advantage of the combination boresight and wire method of determining bearing positions stems from the fact that a reference check point is established midway in the shafting system providing a warning if hull movements during the measurements are sufficient to invalidate the readings. Further, the method facilitates obtaining accurate centers at either ends of struts for boring. The major disadvantage is that shafting must be removed from the shaft line. Properly applied this technique established centers to an accuracy ranging from 1/64″ for short sights to 1/32″ for long sights.

Drops and Openings Method: In the method of drops and openings the relative positions of the flanges of adjacent uncoupled shaft spans are measured. By graphically adding the calculated curves of shaft span deflection to the measurements the bearing positions are established. If a span of shafting has only one bearing, a temporary saddle is added so that each span may be calculated as a simple beam.

Generally the drops and openings technique is applied to the alignment of inboard shafting, although it could be applied any place where flanges are accessible. It has the advantage of allowing the measurements to be taken in a short period of time, minimizing the effects of hull movement. The accuracy depends to a large degree on the geometry of the shaft spans, although bearing positions have been established to an accuracy of $\pm 0.010''$.

In this method the bearing loading (the desired quantity) is not measured directly, but is inferred or calculated from the plot of relative bearing positions. The major disadvantage of this method is the cost of breaking all the shaft couplings.

Bearing Load Measurement: The direct measurement of bearing loads by using calibrated jacks has proven a quick and inexpensive method for repair alignment work. The measured bearing loads are compared with values calculated for the shaft system treated as a continuous beam with all bearings on a straight line.

The method is applicable in all cases where shafting is accessible for jacking, however considerable information is required in advance to permit the necessary calculations. The calculations are invaluable guides in determining acceptable bearing loads and in directing bearing movements needed. Since the variable controlled is measured directly this system is considered inherently most accurate.

Analysis and Remedy: The basis for prescribing corrective action is the correlation of determined bearing positions (or bearing loads), the results of calculations of the shaft system, and all available operational data.

*Fife and Wilbur, "Theory of Statically Indeterminate Structures."

At first impulse it would appear that once measurements were obtained by one of the methods above, corrective action would be automatic. As an example, if the bearings were found by boresight readings to be out of line, the remedial action could be to position all of the bearings so that they would be "in a line". Reflections on the comments made earlier in the paper about hull movements will suggest that such action is folly. For a straight line of bearings on a flexing hull girder is in reality a family of faired curves at different instants of time. Appreciating the above, the practice of accomplishing repair alignment by plotting bearing positions and attempting to prescribe the most easily accomplished bearing movements to result in the bearing positions plotting in a faired curve, has been utilized.

The practice is best described by an illustration of the faired curve plot, Figure 4. Figure 4b shows the bearing positions measured by the boresight method for a particular shaft system in October 1954. Figure 4c shows a similar plot for the same system for data measured in February 1955. Though the bearings are on a straight line at neither of these times, a comparison of the corresponding bearing reactions shows little difference.

Further inspection of Figure 4 will reveal that the thoughtful analyzer has considerable liberty in drawing the faired curve and determining which bearing is to be moved, if any. He is guided in his choice by careful consideration of the operational data and of the peculiarities of the shafting system as revealed by the results of the shaft system calculations.

It should be appreciated that each shaft arrangement imposes its own requirements for accuracy in the positioning of bearings. Where one installation may readily accommodate bearing wear or improper positioning in the amount of 0.100 inches, shipboard measurements of one satisfactory shaft arrangement revealed that lowering one bearing 0.030 inches resulted in an adjacent bearing load increase from zero load to a normal load of 7,500 pounds. It is well to restate that the bearings when positioned on a faired curve will normally accommodate considerable movements of bearings as long as they result in another faired curve. However, random departures of single bearings resulting in a significant departure from the faired curve can result in unloading one bearing and overloading the adjacent bearings.

ILLUSTRATIVE EXAMPLES

The description of several ship repairs follows to illustrate the application of the methods. Prior to examination and discussion, some of the variables influencing alignment are restated for convenience.

a. Arrangement of attached components (Reduction gear, thrust collar, main motor, etc.)
b. Movement of bearings caused by thermal expansion of supporting structure (Reduction gear casing, etc.)
c. Bearing weardown
d. Hull deflection

Ship A

During a post repair sea trial difficulty was experienced maintaining the shaft alley stuffing box watertight. As the ship approached full power, water squirted over the shaft alley space. An un-

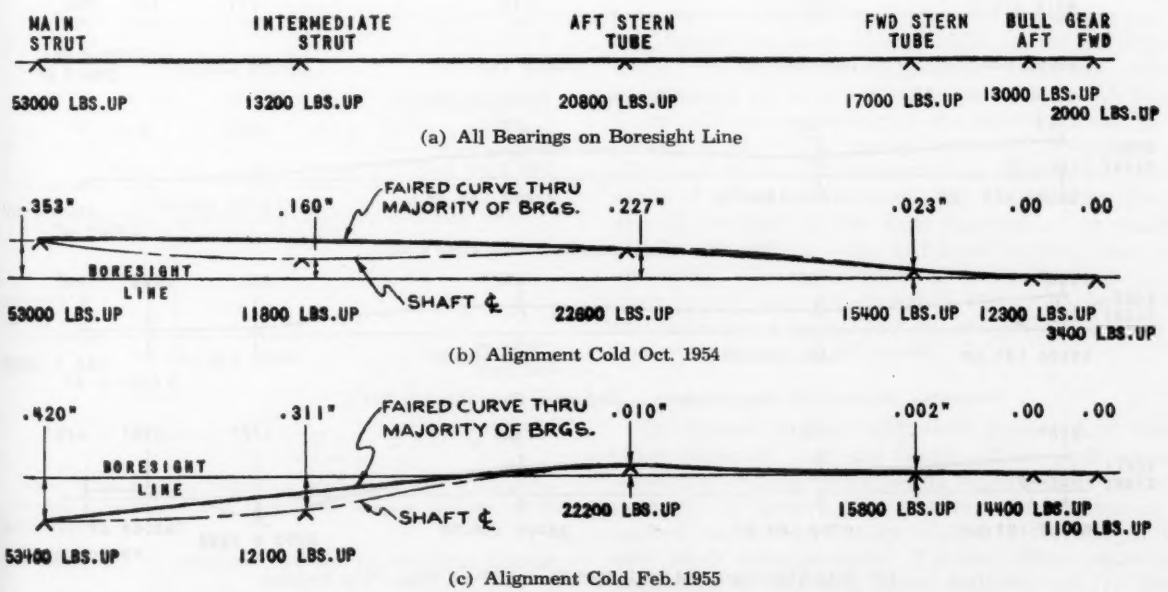


Figure 4. Bearing Loading Changes Due to Hull Deflections—Ship B

Aft Stern Tube Bearing	Fwd. Stern Tube Bearing	Radial Bearing (in thrust brg)	Motor Bearings Aft	Motor Bearings Fwd.
		24100 lbs. up	33000 lbs. up	23400 lbs. up
80200 lbs. up	5200 lbs. down	26000 meas.	32000 meas.	

(a) All Bearings on Straight Line—Aft Stern Tube support at center of Bearing

75200 lbs. up	3100 lbs. up	19700 lbs. up	34800 lbs. up	23000 lbs. up

(b) All Bearings on straight line—Aft Stern Tube support 1 dia. inboard of Aft end

79000 lbs. up	5000 lbs. up		57500 lbs. up	14300 lbs. up

(c) Same as (a) but with Radial Bearing removed

Not measured	13000 lbs. up	13000 lbs. up	22000 lbs. up	Not measured

(d) Measured loading on selected Bearings

Figure 5. Bearing Reactions with Various Alignment Conditions—Ship A.

loaded stern tube bearing was suspected and confirmed by calculations. Figure 5 summarizes the results.

Figure 5a indicates that with all bearings on the boresight line the forward stern tube bearing is unloaded. Since there is considerable bearing clearance, the shaft journal may move within the bearing increasing the shaft column length to the distance between the after stern tube bearing and the combination radial and thrust bearing.

Figure 5a assumes the support points to have been at the midpoints of the bearings. The validity of this assumption for the after stern tube bearing was supported by the measured reactions at the radial bearing and the cock of the after stern tube bearing shown in the boresight plot. Figure 5b shows the reactions resulting if the support point is one diameter in from the after end of the after stern tube bearing.

Elimination of the radial thrust bearing results in bearing loads tabulated in Figure 5c.

The corrective action taken was to raise the forward stern tube bearings 0.045". The measured loading on selected bearings is shown in Figure 5d. The amount to raise the bearing was inferred from calculation and calibrated jack measurements at the accessible bearings. This derangement was precipitated by the effective raising of the after stern tube bearing when the bearing was renewed as part of the repair of corroded and eroded bearing housing and stern tube lands.

It is recommended that studies of this type be undertaken for new designs in order to avoid a bearing loading condition similar to Figure 5a. The sizing of the bearings would thus be more realistic.

Ship B

Ship B experienced reduction gear tooth failures, wiped bearings and reported a rumble at shaft speeds less than 15 RPM. Calculations for bearing reactions for various conditions, later substantiated by shipboard measurements, established that under operating conditions a reduction gear bearing was unloaded. The bull gear reactions attributable to gear tooth loading were combined with static loads to establish the ship's speed at which the journal would bottom. In addition, the static bearing loads were calculated for the uncoupled reduction gear.

Subsequently, shipboard measurements were made to verify the approximated thermal growth of the gear casing, and the unloading of the bull gear bearing. Bearing positions were established by the boresight method and selected bearing reactions were measured using calibrated jacks.

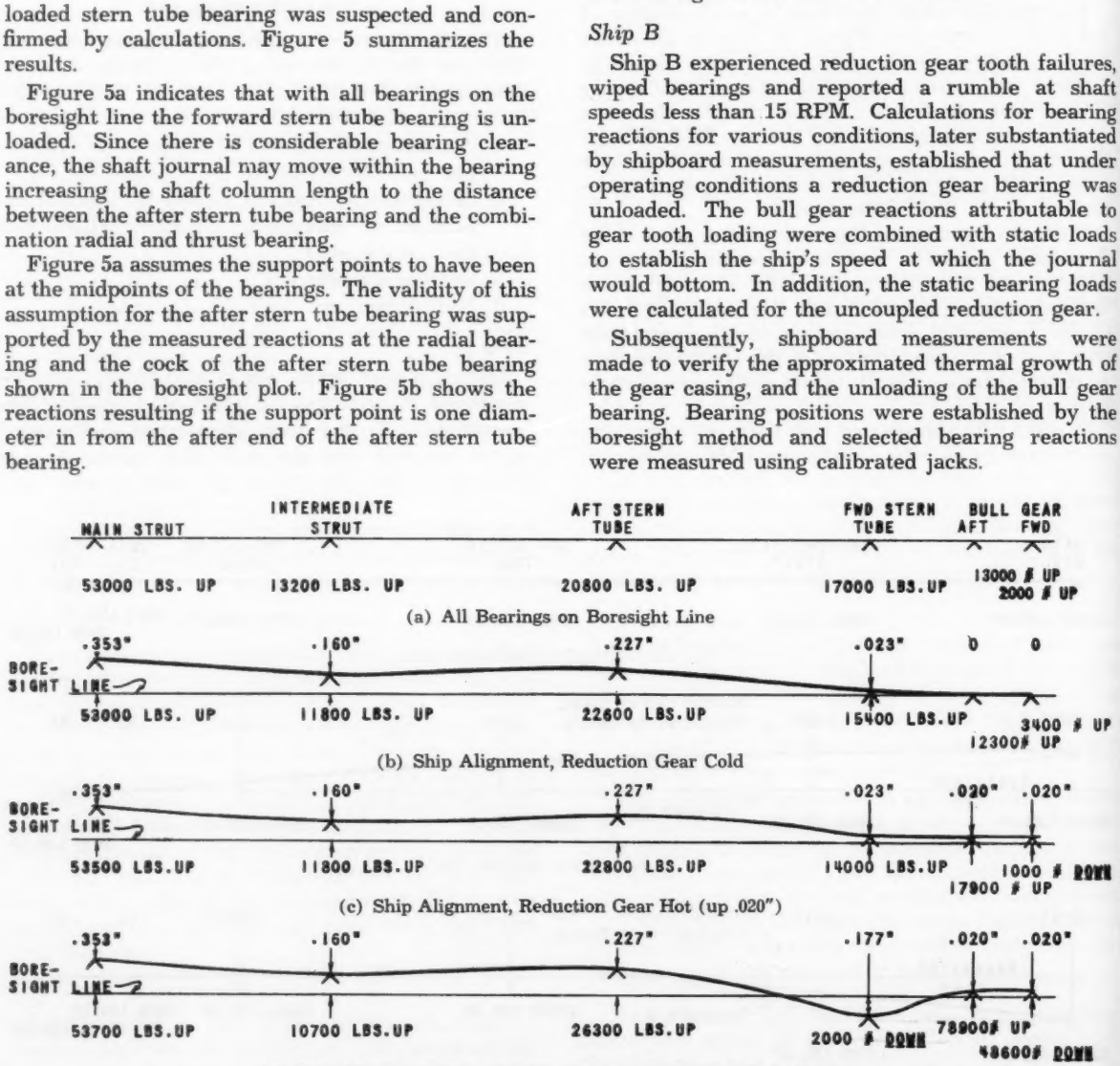


Figure 6. Calculated Bearing Reactions with Various Alignment Conditions—Ship B

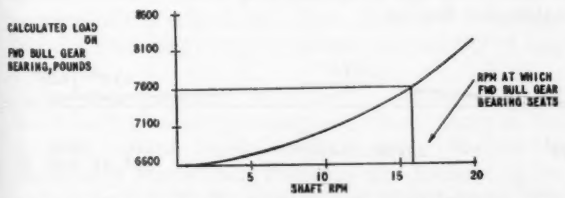
Results are presented in Figures 6 and 7. Attention is invited to the improvement in loading of the forward bull gear bearing by the existing alignment, Figure 6b, as opposed to the straight line alignment, Figure 6a, and to the possibilities for further improving the load split between the bull gear bearings by purposely misaligning the shaft bearings. Figure 6c indicates unseating of the forward bull gear bearing when the gear case was heated. As noted previously this behavior was predicted by calculations and later substantiated by shipboard measurement. Figure 6d illustrates the effect of wear of the forward stern tube bearing. This represents a hypothetical situation since a negative bearing reaction is assumed, and maximum allowable wear was used for the forward stern tube bearing.

Figure 7a compares the uncoupled bull gear bearing reactions associated with the Ship B type reduction gear-thrust collar configuration with another ship having a more desirable load split. Figure 7b shows loading caused as the result of increasing ship speed and illustrates that at about 15 RPM the forward journal could be expected to seat.

The corrective action taken for Ship B was to install a babbit spring bearing in place of the forward stern tube bearing and to position it high to achieve an approximately even load split on the bull gear bearings. The spring bearing permits minor, but significant, adjustments of this bearing's position with accuracy and economy while the ship is afloat. The wear down permitted for a spring bearing approximates the magnitude of wear down which the shaft system can tolerate. It is noted that this bearing replacement did not change the shaft spans, thereby maintaining original satisfactory dynamic characteristics.

SHIP	BEARING LOAD–POUNDS		
	FWD BRG	AFT BRG	THRUST COLLAR LOCATION
B	2977	10345	AFT OF AFT BULL GEAR BRG
E	7380	6932	FWD OF FWD BULL GEAR BRG

(a) Static Bull Gear Bearing Reactions with Bull Gear Uncoupled



Note: When bearing alignment produces a 1000 lb. negative reaction at zero rpm. Bull Gear Fwd. Bearing will seat at 16 shaft rpm.

(b) Shaft rpm. vs. calculated load on Fdw. Bull Gear Bearing

Figure 7. Bull Gear Bearing Loading—Ships B and E

Ship C

In planning for the replacement of shafting in Ship C, investigations of the proposed shafting arrangement, Figure 8, revealed extremely sensitive conditions would prevail in positioning the forward stern tube and the spring bearing relative to the bull gear bearings.

Figure 8b shows that the forward bull gear bearing would unload when the gear casing expanded due to heating. To equalize bull gear bearing loads would require raising the shaft alley spring bearing 0.022″ as shown in Figure 8c. In this condition only 0.022″ weardown would be required of the forward stern tube bearing to unload completely the after bull gear bearing, Figure 8d. The limits of vertical positions for the forward stern tube bearing are shown in Figures 8e and 8f assuming a minimum of 2000# is acceptable for the loading of either bull gear bearing.

From the above it is apparent that the bearing arrangement would permit approximately 0.034″ weardown of the forward stern tube bearing if the bearing could be installed at the exact optimum position. Such fine working tolerances are difficult to achieve. Also the rather large weardown permitted in water lubricated bearings could not be realized in this installation.

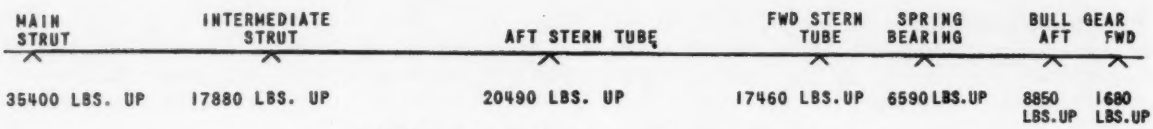
If the shaft alley spring bearing is eliminated and a babbit bearing substituted for the forward stern tube water lubricated bearing an arrangement is achieved which is more easily installed and maintained. Results of these studies are shown in Figures 8g and 8h.

Ship D

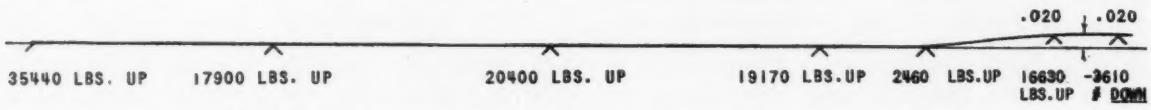
In the case of Ship D, a condition of a whipping shaft concurrent with severe rubbing of bulkhead stuffing boxes was evident. In addition, visual observations indicated the presence of a shafting system lateral vibration critical. The results of shipboard investigations conducted underway are summarized in Figure 9. An examination of this figure indicates the existence of a lateral critical frequency at 1110 CPM excited by propeller blades at 370 shaft RPM. The deformation pattern for this mode of vibration indicates that the existing fore and aft locations for the shaft bearings are at nodal points. In addition, the bulkhead stuffing box is anti-nodal. The installation of two additional line shaft spring bearings, shown in Figure 9, removed the bothersome lateral critical and permitted satisfactory operation at all speeds.

CONCLUDING REMARKS

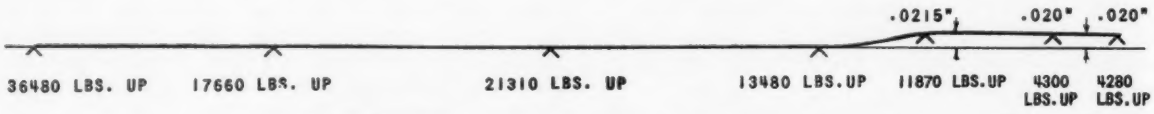
The system of shaft alignment presented in this paper together with the results of its application constitute current practices as experienced by the authors. It appears that the type of studies described could be profitably applied in the design of new shaft arrangements. Further effort appears warranted in obtaining better definition of (1) the significance of the various bearing movements on the



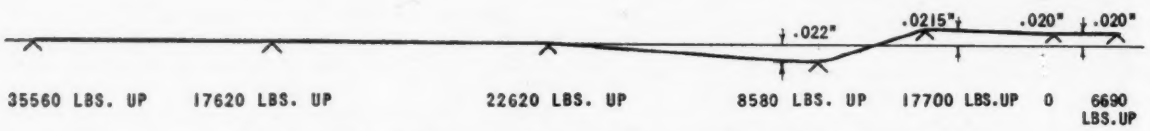
(a) All Bearing on straight line afloat and Gear cold



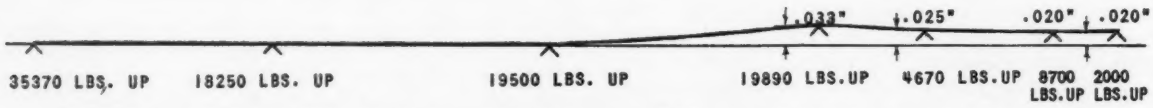
(b) All Bearing on straight line, Reduction Gear hot (Bull Gear Bearings up .020"



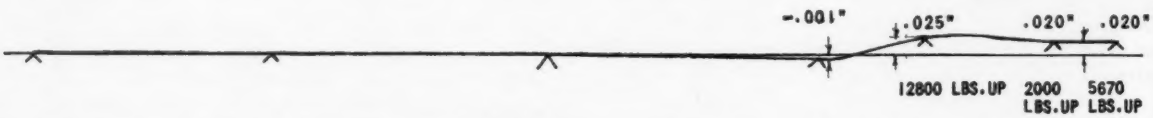
(c) Spring Bearings raised to give equal Bull Gear Bearing reactions



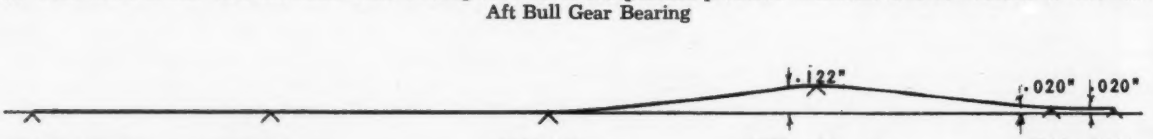
(d) Wear down of .022" on Forward Stern Tube Bearing unloads Aft Bull Gear Bearing



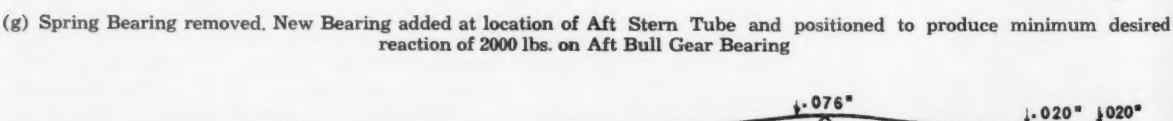
(e) Position of Fwd. Stern Tube Bearing to produce minimum desired reaction of 2000 lbs. at Fwd. Bull Gear Bearing



(f) Wear down of .034" of Fwd. Stern Tube Bearing from the above position produces minimum desired reaction of 2000 lbs. at Aft Bull Gear Bearing



(g) Spring Bearing removed. New Bearing added at location of Aft Stern Tube and positioned to produce minimum desired reaction of 2000 lbs. on Aft Bull Gear Bearing



(h) Position of new Bearing set to produce minimum desired reaction of 2000 lbs. on Fwd. Bull Gear Bearing

Figure 8. Calculated Bearing Reactions for Various Alignment Conditions—Ship C

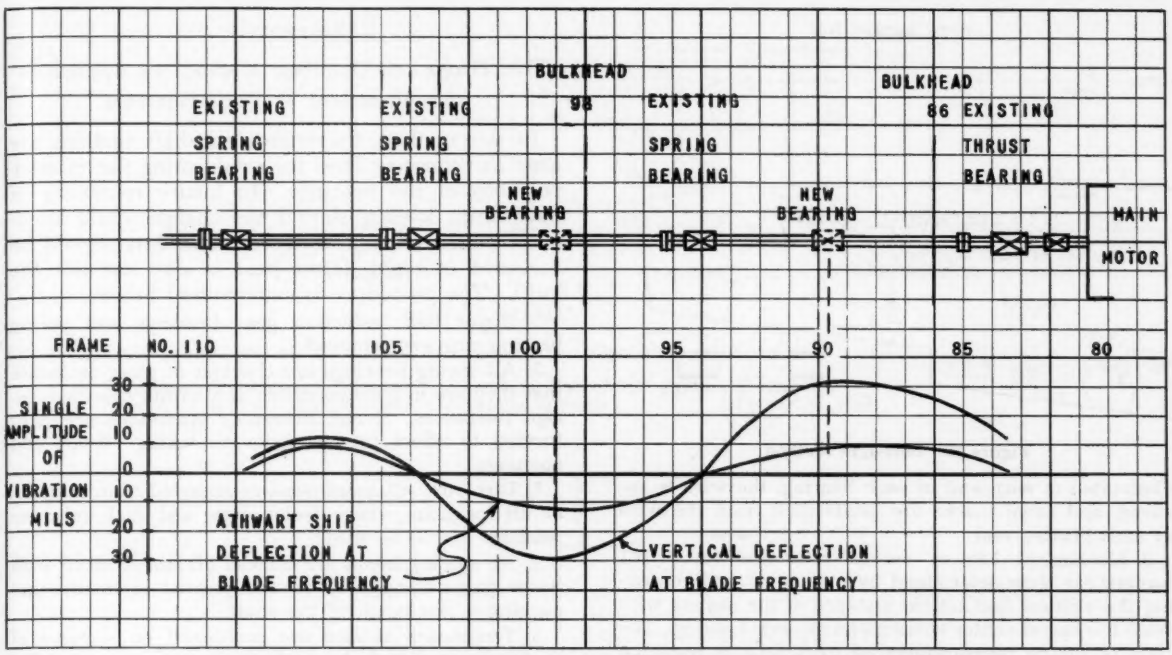


Figure 9. Dynamic Deflection Patterns of Starboard Shaft at Critical Speed—Ship D

shaft system, (2) reexamination of the advantages of the familiar stern tube bearing configuration common in ship design and (3) a more economical and accurate technique for measuring bearing positions (or loading). Towards the latter end, the method of direct measurement by jacking is being developed. It is conjectured that the existent bearing reactions revealed by this analysis may affect bearing sizing practices and shaft stress analysis.

ACKNOWLEDGEMENTS

To many persons, who were actively engaged in alignment work at the Boston Naval Shipyard in years past, the authors express their appreciation for the legacy of information they contributed to this paper.

Particular appreciation is due Commander S. V. Wroblewski, USN, Design Superintendent, for the encouragement given, without which the paper would not have been written.

The contributions of Messrs. R. E. Wolfe and E. M. Lindholm, Naval Architects, Design Division, in accomplishing calculations and shipboard investigations of shaft alignment problems are worthy of special note.

For their repeated assistance in problems in the mechanics of accomplishing alignment changes, the authors extend thanks to Messrs. A. Somerville, Machinery Supervisor, Planning and Estimating Division, and J. F. Travers, Chief Quarterman Machinist, Production Department.

APPENDIX A

The Combined "Boresight and Wire Method" as Applied to the Alignment of Shaft Bearings

The method is basically a physical means of extending a reference line or base (such as the line established by reduction gear shaft) by the use of a telescope and a wire. Bearing positioning is referenced to this line. The procedure is summarized below:

1. With all shafting cleared from the line of sight, the boresight equipment is secured firmly to the after flange of the reference shaft (bull gear, main propulsion motor, or after line shaft). The telescope is aimed in the direction of the target.

2. As the shaft is revolved, the scope projects a conical surface whose axis coincides with the centers of the rotating shaft. These lines of sight are projected on a target mounted in the after end of the after-most bearing. The radial center of these points establishes the position of the reference line. This reference point is checked by repeating the above procedure with the scope adjusted to project points on a larger radius.

3. By means of hermaphrodite calipers, the physical center of the end of the strut barrel is determined and marked on the target, and a plumb line is drawn on the target for a vertical reference.

4. Proof marks are established on the after end of the strut barrel from the centers projected by the telescope.

5. Steps 2, 3, and 4 are repeated for one end of another bearing about midway between the telescope and the target as previously described. (This data is used as a check point).

6 The telescope and the target are removed and a wire is strung from the mechanical center of the flange on which the telescope was mounted through all the bearings. The wire is centered to the proof marks of Step 4 above and supported entirely from the ship.

7. A transit is set up on an adjustable mount abaft the aftermost strut and picks up the line of the boresight center in the after strut and the center of the flange from which the wire runs. (The flange on which telescope was mounted.) Offset targets are used for this work.

8. With offset targets and straight edges, the sag is

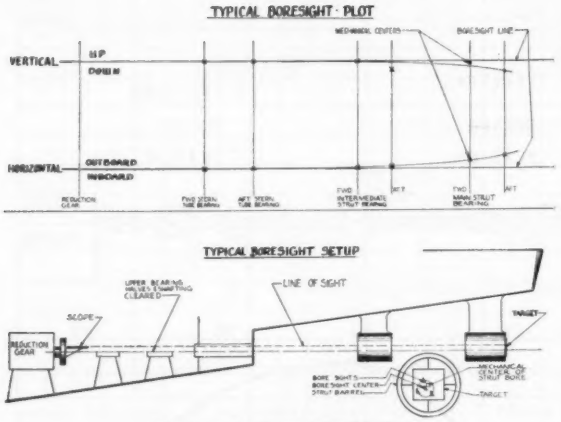


Figure 10. Boresight Method

eliminated at each end of each bearing, the wire is secured, and proof marks are established from the wire at each bearing end.

9. Mechanical centers versus reference (or wire) centers are then determined by measuring and recording the vertical and lateral distance of the sagless wire from the barrel center at each end of each bearing. The time of taking measurement at each face is recorded along with the readings.

10. At this point, the data of Step 5 is checked with the data for that point from Step 9. If close agreement does not occur, the boresight work is repeated until satisfactory agreement is reached.

11. The comparison of boresight versus mechanical centers is then plotted. Using a convenient scale, the fore and aft positions of the bearings are plotted on the abscissa, and the mechanical centers are plotted above or below a straight (boresight) reference line as ordinates. Illustrative data is shown on Figure 10.

APPENDIX B

The "Drops and Openings Method" as Applied to the Alignment of Shaft Bearings

In this method, the uncoupled shaft sections are used as reference lines for establishing the relative positions of the bearings. To insure reliability of data, ships loading should not change while alignment measurements are made, and data should be recorded at night (after heat of day has left the hull). The procedure is summarized below:

1. Upper half reduction gear bearings and spring bearing caps are removed.

2. All spring bearings are checked in place to insure that they are in good condition and within specified design tolerances. If not, necessary operations are performed to effect repairs, prior to taking "drops and openings".

3. Dirt, rust, oil, paint, etc. are removed from surfaces at all locations where feeler gage and dial indicator readings are to be made.

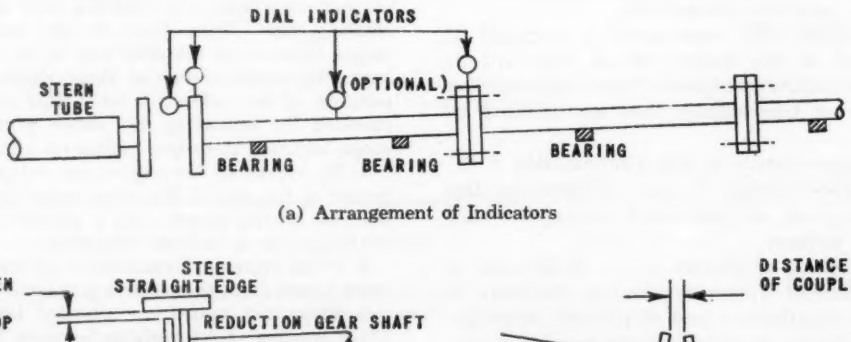
4. All stuffing boxes are slacked off and checked with feeler gages completely around journal to insure that packing is not touching the shaft.

5. Temporary saddles are positioned on sections of shafting supported by only one bearing. (Saddles equipped with rollers may be required for shaft runout check).

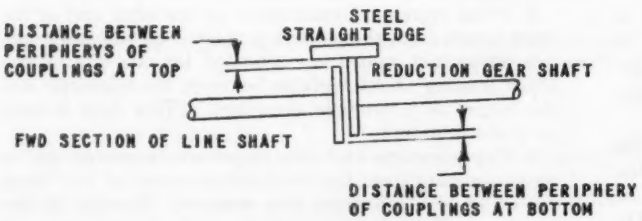
6. The following procedure is used to determine that each section of shafting is straight and flange faces square:

(a) Aftermost inboard coupling is broken.

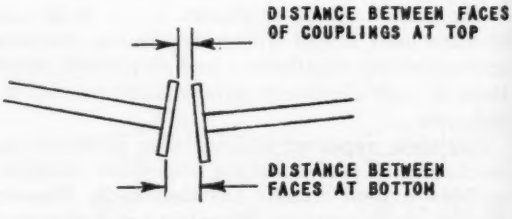
(b) Dial indicators are mounted on both coupling flanges of last section of inboard shafting. These are set on the periphery and in the same plane. Another indicator is mounted against the coupling face or after



(a) Arrangement of Indicators


DISTANCE BETWEEN PERIPHERYS OF COUPLINGS AT TOP
STEEL STRAIGHT EDGE
REDUCTION GEAR SHAFT
FWD SECTION OF LINE SHAFT
DISTANCE BETWEEN PERIPHERY OF COUPLINGS AT BOTTOM

(b) Method of measuring Drops

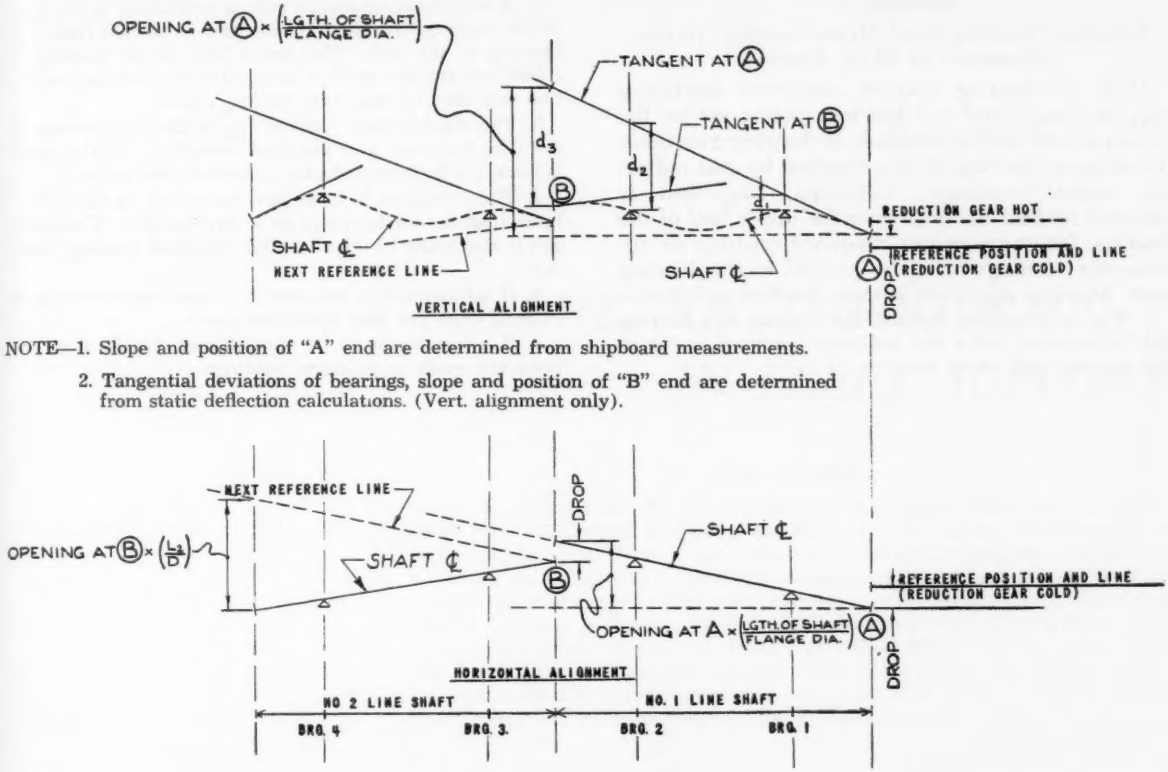


(c) Method of measuring Openings

GENERAL NOTES

1. Openings are differences between top and bottom of inboard and outboard feeler readings of flange faces.
2. Drops or rises are the average of feeler readings between the coupling peripherys at the top and bottom or inboard and outboard edges of the flange couplings.

Figure 11. Method of Taking Data for Drops and Openings



NOTE—1. Slope and position of "A" end are determined from shipboard measurements.

2. Tangential deviations of bearings, slope and position of "B" end are determined from static deflection calculations. (Vert. alignment only).

Figure 12. Drops and Openings Plotting Techniques

flange. (see Figure 11). If the dial indicator on the flange periphery shows a runout of 0.002″ or more, runout is also measured at the midpoint of that length of shafting.

(c) The shaft is rotated through two revolutions and simultaneous readings of the dial indicators are recorded.

(d) Progressively work forward toward the reduction gear, breaking each coupling in turn and indicating flanges for each section of line shaft as described in step (b).

7. With each section of shaft supported at two points, and flange couplings parted sufficiently to allow free movement between adjacent coupling flanges, measurements are taken at each coupling to determine the relative "drop" (or offset) and "opening" (angular change), between adjacent flange couplings. The shipboard measurements are illustrated in Figure 11. With two teams measuring and recording all data, a check is obtainable. Facial readings are the more critical and the openings obtained from these readings should check within 0.002″.

8. The relative positioning of the bearings from a reference line (such as the line established by the reduction gear) is determined graphically. The plotting technique is illustrated by Figure 12. "Opening" is defined as the difference between top and bottom, or inboard and outboard feeler readings between flange faces; and "drop" (or offset) as the relative displacement of the flange coupling (vertical or horizontal) as determined from readings at the coupling peripheries.

(a) Referring to the plot of horizontal alignment data (Figure 12), the center of the forward end of the No. 1 line shaft section is positioned by the "drop" reading. The opening multiplied by the ratio of shaft length to flange diameter positions the after end of the shaft. A line drawn between these points will pass through the bearings (or supports) for this section of shafting. This line and the position of the center of the after coupling flange of the No. 1 line shaft are the reference position and line for plotting the data recorded at the coupling between No. 1 and No. 2 shaft sections.

(b) Relative to plotting of vertical alignment data, the shafting deflects under its own weight and the bearings do not lie on a line perpendicular to the coupling flanges. Data for the tangential deviations of the bearings and shaft ends must be computed for the shaft section as a simple beam with supports as positioned during recording of shipboard measurements.

(c) In plotting the vertical data for No. 1 line shaft, the position of the forward flange and a line perpendicular to the flange coupling is drawn in the same manner as described for horizontal alignment. This line is labelled tangent at A in the illustrative plot. The tangential deviations of the bearings and the aft end of the shaft are layed off to position these points. The method is repeated for the next section of shafting; the position of the center of the after coupling flange of the No. 1 line shaft and the tangent to the elastic curve at this point (labelled tangent at B) are the reference position and line for plotting the data recorded at the coupling between No. 1 and No. 2 shaft sections.

APPENDIX C

Utilizing "Bearing Load Measurements" in the Alignment of Shaft Bearings

With all shafting coupled alignment conditions may be established and bearings positioned by the measurement and evaluation of bearing reactions. Trueness of shafting is first checked by dial indicator runout readings. Horizontal alignment is checked by lead readings over the upper half of the bearing, feeler gauge side clearance readings or micrometer readings from the journal to the bearing seat. Vertical alignment is then checked as follows:

1. For each bearing desired, the top cap and bearing half is removed and a dial indicator mounted on top of the journal, and set at zero.

2. A calibrated hydraulic jack is positioned at the face of the bearing. The load of the shaft is shifted from the bearing to the jack. The lower half of the bearing is rolled out and the shaft is lowered to its free sag condition and the dial indicator reading noted.

3. The shaft is then "jacked" up to the zero setting of the dial indicator and the load recorded. As the shaft is raised, a loading deflection curve is also made.

4. The measured load is then compared to the calculated load for all bearings on a straight line. Comparison is also made to the allowable designed bearing pressure.

5. If adjustment is necessary, proper repositioning is evident from the load deflection curve.

6. If a bearing is to be repositioned, check observations are made at adjacent bearings.

COMMANDER J. H. GARRETT, SC, USN

A SCHEME FOR IMPROVEMENT IN FLEET SUPPORT

THE AUTHOR

was born in Richmond, Va. He attended public schools in Richmond and graduated from the College of William and Mary. In 1942 he was commissioned an ensign in the Naval Reserve and served as a line officer with the Amphibious Forces in the campaigns in the North, South and Western Pacific. In 1946 he transferred to the regular Navy as a lieutenant in the Supply Corps. Since that time he has been stationed at the Marine Corps Air Station, Cherry Point, N.C., was supply officer of the USS Kula Gulf (CVE 108), has been assigned to the staff of ComServLant and was a student at the Naval War College, graduating in 1953. Commander Garrett is now a member of the staff of the Naval War College in the Department of Research and Analysis.

As our Navy moves ahead in this age of technology, it becomes more and more dependent upon complex equipment for its might. At first reading this sounds like a rather trite statement. The full implications of this are not so trite, especially from the standpoint of the logistician. His ability to maintain a timely, responsive and efficient logistic support system will not only call for more ingenuity, but the support system for our sea forces will be all the more essential.

In the past, logistic support to varying degrees has rested upon the American talent of improvising. While such improvising has for ages been a hallmark of good ships, we are now rapidly entering a new age. We should not now belittle improvising, but it is certainly not the gimick it once was. More planning today for less improvising tomorrow will save many frustrating experiences later when improvising just will not work.

To illustrate the implications of the growing technology, visualize the problem of a burnt out radio coil twenty years ago. A radioman worth his salt could rewind it with standard wire carried in the GSK storeroom. Repairing some of today's simplest components of our shipboard equipment is far beyond the skills and the equipment found aboard ship. Then too, the lack of the radio, though a decided inconvenience, would not have necessarily made the ship ineffective twenty years ago. Today though, the lack of fully operational equipment can seriously jeopardize a ship or a task force. It sometimes seems that the masters are becoming the slaves of the equipment, rather than vice versa. However, this situation does point up the fact that unless we keep these complex slaves working and under our control they may, in truth, become Frankensteins.

The electronic tube is probably today's classical example of the precision made repair part, but the aray of parts of about equal complexity in our combat aircraft is sometimes glossed over. Yet, we should be taking our cue from this technological achievement. It could well be that the airplane is showing the way of the future in the resupply problems of its mass of complex repair parts. As technology slowly revolutionizes our ships as it has the aircraft, the repair parts problem in areas which have never given trouble could erupt into nasty boils.

Logisticians, faced with their charter of sustaining combat forces in the highest possible state of combat effectiveness, must be quick to learn lessons from the past and to detect symptoms of impending trouble.

With this mission in mind this paper was prepared for two purposes. The first was to propose in broad terms a system for one segment of fleet logistic support. The other was to create provocative thought on this subject.

Three interrelated proposals are discussed in sequence:

(1) A means for improving the self-sufficiency of our combat ships through a new concept of loading.

(2) A method of keeping a ship's load of repair parts fully ready for use.

(3) A proposal for rapid resupply to task forces at sea.

SELF-SUFFICIENCY OF COMBATANT SHIPS ON DOWNWARD TREND

For many years of naval history ships of the line could sail so equipped and supplied that they could operate completely self-sufficient for extended cruises. Only occasional stops were necessary to take on water, fresh provisions, and infrequent naval stores.

When steam replaced sail, the naval logistic problem received its first real increase in magnitude. The coming age of technology was making its first major impression since the discovery and application of gun powder, and the advent of iron hulls. Still, however, war ships maintained a high degree of self-sufficiency, only adding the problems of coaling and simply designed repair parts. This development also marked the emergence of an embryonic mobile logistic support concept as colliers were added to the Fleet. Even so, fixed bases appeared to have received more thought as a means of supporting deployed fleets.

As technology began to have increasing influence, combatant forces became less and less self-sufficient. To meet this challenge the concept of mobile logistic support began to take on more definitive form. Much of the concept's initial impetus can be traced directly to the development of fleet oilers to satisfy the fuel hungry ships. Technology continues to nourish the concept of mobile support and force its growth more than any innate momentum in the idea.

At this point it seems wise to take a questioning look at our concept of mobile logistic support and try to determine if we are still on the right track. Are the guide lines adequate? Are we getting the most from the concept? Are bolder developments necessary to meet the challenge of tomorrow's Navy?

As the self-sufficiency of the combatant forces has decreased, increased dependence upon the mobile logistic support force has become necessary. As a result, the logistic forces, in addition to a number of other services, must maintain a supply inventory approaching one hundred thousand different stock numbers. Even so, this force can only supply upwards of 80% of the items requested by Fleet units. Indeed the announced goal is only 90%. Beyond that goal the relationship of inventory levels to effectiveness in filling requisitions gets completely out of sensible proportion.

For example, it has been estimated that potential requests for ship repair parts approximate 5,000,000 different items. However, with only 200,000 items in the ship parts supply system, about 90% of the requests received from the Fleet can be filled. The other 10%, being scattered over better than four and a half million items, are procured directly from industry or higher assemblies are furnished. In the stores issue ships of the mobile logistic support force about 15,000 ship repair parts items are carried. This inventory level provides an issue effectiveness of 60%–70%. It takes no statistical genius to see that the point of diminishing returns in logistic effort can be quickly reached by trying to carry large inventories of slow moving material in forward areas either afloat or ashore.

With electronic and ordnance repair parts the picture is similar, even though less severe. Even with general stores, a similar statistical picture exists.

Yet, ships and aircraft continue to need (as opposed to desire) the items not carried in the mobile logistic force.

As we become more and more dependent upon technology to wage wars, it becomes increasingly apparent that some means must be found to increase logistic self-sufficiency of our deployed forces (or at least stop the trend of decreasing self-sufficiency) and to improve our logistic system to supply needed items, or a combination of both.

CHARACTERISTICS OF THE USAGE OF NAVAL SUPPLIES

To start, let's take a look at the characteristics of the material requisitioned by ships. In line with the requisition pattern previously observed, it has been determined that only a small number of different items account for a decided majority of the resupply of repeat items to the Fleet. In broad terms they are the housekeeping items (rags, toilet paper, soap powder, stationery articles and the like), provisions, fuel, ammunition, clothing and certain electronic tubes. These items, characterized by a high percentage of repeat orders, can be narrowed down into a list of less than 2,000 items, the specific number depending upon how far you want to go into the "grey" area. As has been noted, the balance of the Fleet's demand for supplies is spread over hundreds of thousands of items.

A decided characteristic of the "fast moving" items is that either individually and/or collectively they are large cubic volume items. Again, in general terms, the slow-moving items are small cubic volume items. As an example, the fast-moving general stores items account for about 80% of the space occupied by the supply of general stores items carried aboard a fleet issue ship (AKS).

For a number of years standard logistic instructions have directed balanced loads of stores for combatant ships, expressed in terms of specific days of endurance. In some instances this has led to judging inventory efficiency by dividing quarterly monetary expenditures into the value of inventory on hand. This, in theory, would provide a specific "days of supply" on hand. Although this approach may have variable validity in measuring an inventory of parts to support a production line or of rations to feed a specific number of men, it does not

appear to have much meaning when applied to an inventory to support a repair and maintenance program. Certainly, these recently determined characteristics of the Navy's issue rate do not support such a procedure for determining days of supply on hand.

I bring this point in to show how closely our thinking has become tied to a concept of balanced loading of combatant ships. At least to some extent, I believe that this has been fostered by one technique that has grown out of a present application of the mobile logistic support concept—the regular cycle of replenishment at which time various categories of material are brought up to predetermined levels of supply.

Without being specifically aware of what they are doing, some combatant ships, either consciously or unconsciously, carry an unbalanced load of certain slow moving items. If space for small volume, slow moving items is available, very often the inventory level of these items is increased. So far, this has been an uncoordinated procedure, having many drawbacks. However, if it could be coordinated, definite benefits immediately suggest themselves.

Since ships, even in wartime, must be overhauled sometime, the possibility of loading ships with slow moving, small cubic volume items in sufficient quantities to last from overhaul to overhaul immediately suggests itself. Under such a procedure, it is not contemplated that an attempt be made to load so that the last gasket on board is used as the ship's bow enters a dry dock. Indeed, in consideration of the issue characteristics we have previously observed this could not be done with any degree of accuracy.

However, in the light of previous usage data, breakdown history, and engineering judgment, it is considered practicable to specify a load of supplies, which have the characteristics of low usage and low volume, which should be of reasonably high reliability to last a ship from overhaul to overhaul, especially if backed up by a sensitive resupply system similar to the one I will describe later in this article. As a safety factor and to cover such contingencies as delays in the overhaul schedule, increases probably should be added to any computation made on the basis of a normal overhaul cycle. To take care of erratic or unexpected demand and to prevent trying to slice the cake too thin a low limit or reorder point should also be indicated in any stocking list for a ship. It is also obvious that so-called insurance items, (only one is stocked aboard a ship) should be re-ordered when used. (Bear in mind that this concept does not visualize stocking a minimum of one of each potential repair part on each ship. A judgment factor considering many facets will enter into the preparation of such a list. Means of rapid delivery of requirements, as discussed later in this paper, will be an essential consideration when preparing such a supply list.)

Such a concept as this has been given a characteristic name—endurance loading.

With the adoption of such a scheme, the nuts and bolts of mobile logistic support can be greatly reduced. Since ships would not require routine replenishments of low usage items, it would not be necessary to carry them in the mobile logistic support force. Consequently the different number of stock items carried in the mobile logistic support force could be reduced from around 100,000 to about 2,000.

It takes little imagination to see the tremendous savings of effort and manpower that can result in just two areas:

(1) The mass of paper work in the Fleet associated with requisitioning will be reduced.

(2) Logistic effort in the support forces which is devoted to inventory control can be concentrated on fewer items, resulting in greater efficiency in stock upkeep of these items.

In addition to this savings in the Fleet, the substitution of wholesale issues to the Fleet (at overhaul periods) for a constant stream of "retail" issues (as is now the case) should result in a great workload savings in the Navy Supply System ashore.

All in all, this aspect of the scheme will help keep the logistical burden of our Navy close to manageable size.

LOCATION, IDENTIFICATION AND CONDITION OF REPAIR PARTS MUST BE TOPS

It follows that the increasingly vital repair parts aboard ship under this concept must be capable of being speedily located and quickly identified, and be in a fully acceptable material condition.

The recently inaugurated bin-drawer stowage system for shipboard repair parts is a big step forward in meeting the first criterion. Present conditions in the Fleet (and we should not expect any great changes for the better any time soon) are not very favorable for maintaining the second criterion. The aggregate of constantly changing stock numbers, constantly changing allowances of repair parts resulting from added and deleted equipment, the inevitable human errors in marking or tagging material, and the mounting administrative burden on military personnel afloat make this a supply and engineering officers' near nightmare.

The extent of the identification problem is typified by a few recent statistics at the Naval Supply Center, Norfolk. Around 50% of the requisitions submitted by the Fleet for machinery and electrical repair parts either list the repair part under an obsolete stock number or without any stock number, depending in this latter case upon the Center's technicians to arrive at a stock number from whatever descriptive data the ship can dig up. The technician payroll cost at the Center and the delay in supply service as a result of this condition is something to reckon with.

An apparently characteristic phenomenon of today's technology is that of preciseness. Consequently, repair parts to support today's and tomorrow's equipment must possess this same characteristic. Dirt, moisture, handling and indeed sometimes just

a period of lying on a shelf are deadly enemies of accurately tooled fittings, close tolerances, and delicate connections. In the face of this, the most careful attention must be given to the material condition of repair parts aboard ship.

Considerable concern has been expressed about the condition of material in store. In certain categories of technical parts this can become a serious problem throughout the system. The condition has resulted from a number of reasons, not the least of which is simply Father Time. Certain steps have been taken to combat this particular menace aboard ship, such as publishing lists of material with specific shelf lives and the exchange of certain items carried in store aboard ship for a specified length of time for "newer" stores stocked ashore.

Even so, occasions still arise when suddenly it becomes known that particular items scattered throughout the bins and repair parts boxes of combatant ships are unreliable for use as repair parts. Not infrequently directives are sent to all units of the Fleet, directing the disposal of material as being unfit for use, or establishing a recovery or revamping program for specific items or categories of items.

Still though it is the general policy to place repair parts aboard ship at commissioning and let them lie on the shelf or in a repair parts box, under practically every known storage condition, until withdrawn for repairing a piece of equipment. To find at that date that the repair part is unfit for use is much too late. Yet it continues to happen. The sensible course of action, it seems to me, is to take positive, periodic action aboard each ship to preclude the situation of carrying defective repair parts aboard ship.

OUR TECHNOLOGICAL NAVY CALLS FOR BOLDER MEANS OF SUPPORT

Bolder, more imaginative programs are overdue to meet the challenge of tomorrow for a more dependable shipboard supply program. So far it has been piecemeal assistance by the shore establishment. The big burden for supply readiness rests squarely on the stooping shoulders of afloat personnel—sent to sea primarliy to learn to fight the ship effectively, but in the end so burdened with administrative matters and "ship's force work" that it sometimes appears that military matters are only to be considered as incidental.

In the slower tempo, pre-World War II Navy, supply and engineering personnel could take care of the repair parts problems almost in their spare time. Relatively, there were not very many parts to worry about and equipment repairs were far simpler than they are today.

Without a work measurement program for shipboard use, a statement that ships, without positive help from the shore establishment, must and can insure that repair parts at all times will be speedily located, quickly identifiable and in top material condition, can hardly be proved or disproved. I believe that I stand on solid ground, however, when I say that the decided majority of officers who are familiar with this problem will say that all ships need considerably more help in this steadily worsening situation.

The cyclical overhaul period, when a ship's bottom is being scraped and repainted, when its equipment and machinery are being overhauled or replaced, and as the ship in general is being physically rejuvenated, is the logical time to give the same treatment to its technical stores. In a very confused manner this is now done to some degree, when repair parts are juggled during overhaul periods to bring the supply levels into consonance with newly installed equipment. But as every supply officer, who has had experience with this choice little task will testify, this is a highly unsatisfactory system, and it gets worse as the years roll by and the list of specialized equipment and supporting repair parts grows longer and longer.

The only way that has come to my mind to lick this problem and reach some sort of assurance that repair parts can meet the aforesaid criteria of capable of being: (1) speedily located, (2) quickly identified, (3) materially ready, is to offload *all* technical repair parts at overhaul. They would then be processed through a shore based activity skilled in identification and determination of their fitness for issue. While this is going on, modern repair parts stowage facilities should be installed aboard the ship (if not already aboard). The shore based activity should then restock the ship with parts which are tailored to the new inventory of shipboard equipments and which are properly identified and fully ready for use and in accordance with the endurance loading concept, previously discussed.

It may be asked why I look to the naval shore establishment to do this work. If history is any sort of guidance for the future, I think that it is perfectly obvious that such a program has become too big for the ship's force during an overhaul period.

The reasons for this are most apparent. The ship's overhaul period is the time for large personnel turnovers, sending men to training schools and granting much deserved leave. In addition the ship's company must perform continuing military tasks such as security patrols, fire watches, and regular inport watches. All of this time, the normal administrative burden of a commissioned military unit continues—the mail must be sorted, routed and acted upon; letters must be written in the presence of the acrid fumes of an acetylene cutting torch; the division officer must discuss personnel problems in the din of chipping hammers; new men must be oriented into the ship's organization; and sweepers must still man their brooms. None of this situation is conducive to the specialized and meticulous process necessary to meet the criteria of insuring that repair parts are properly identified and in first class material condition. To expect ships to continue to perform this growing and tedious task is likened in my mind to giving arbitrary and petty orders, withholding

the means for their accomplishment, and then punishing the junior for disobedience.

The experience of the shore establishment loading tenders and repair ships in this manner proved highly satisfactory, especially with ordnance repair parts.

Do not get me wrong here, though. I by no means advocate that ships should be relieved of all of their traditional responsibility for their own material readiness. Such a shift in responsibility would lessen our mobility and self-sufficiency even more. Without any apology for our sea forces, I simply want to focus upon the vital need for a still more logical division of labor between the shore establishment with its highly specialized skills, and the ships with their more generalized and military proficiencies. The present wide-spread practice of yard availabilities of various sorts must be extended to a "supply availability" at a shore activity possessing the proper skills for this function.

Upon completion of this processing of the repair parts by the naval shore establishment and after they are loaded aboard ship, we would have a ship really ready, supplywise, almost—

STILL ANOTHER HOLE NEEDS TO BE PLUGGED

We are still faced with a problem. How can those items not carried in the mobile logistic support force or aboard individual ships be furnished systematically and expeditiously to ships on the firing line? Weight and space, not to mention the momentary and production considerations, preclude stocking either on each ship or in the mobile logistic force all items to replace every potential repair part in the force. Equally important, how can needed critical items be expeditiously moved to the operating units without costly and long logistic "pipelines?"

We have right now within the Navy the means to condense time and space factors and provide the sensitivity needed to provide the supply support needed by the fleet. This is in the Navy's air arm. By more fully exploiting its existing capabilities, a rapid transportation link from rear area supply points to task forces at sea can be achieved. This would involve the flying of the requested material from CONUS points to an airfield overseas in the vicinity of the task force (as currently conceived). Here the material would be transferred to a cargo type aircraft, capable of landing on an aircraft carrier type deck of a ship of the underway replenishment force. (Specially configured S2F's and AD's are now being considered for this purpose.) Let's call this ship a CVUK (carrier-stores ship).

Aboard the CVUK the fleet freight would be sorted and further delivery effected to the requesting ship by means of its detachment of helicopters during replenishment operations, or in extreme emergencies flown from the CVUK to a CVA on the line for either its own use or for further transfer to a ship in company.

As a routine, it is considered to be neither desirable nor feasible to saddle a combat carrier, as was done during the Korean War, with the task of being an air freight transshipment depot, although improvisation in extreme emergencies would be practicable. Continuing this improvization type operation will become more infeasible as our dependence upon technology increases and this type of rapid supply support will be required more and more for ships other than aircraft carriers.

The CVUK could serve many other purposes to further add to its value as a member of the logistic support forces. It could be a stores issue ship, carrying either provisions, the fast moving items previously discussed, act as an oiler (until nuclear power and/or more efficient fuels radically diminish this problem), receiving its replenishment from the remnants of conventional fleet oilers prior to their return to a reloading point, or a combination of all of the above—a one stop replenishment ship.

Other tasks could also be performed by the CVUK:

Operate a guard mail center.
Operate a post office for the force.
Operate a movie exchange.
House a pool of highly skilled technicians to assist individual ships with repair problems during underway replenishment periods.
Operate an auxiliary photographic lab to supplement those on combat ships.
Serve as a hospital for emergency cases and act as an evacuation point.
Act as a limited personnel receiving ship, carrying spare air crews, as an example.

The versatility of the helicopter would greatly increase the efficiency and reduce the time necessary to perform these tasks and those of transferring personnel and material rapidly during a replenishment operation. The diversion of helicopters from combatant aircraft carriers for this purpose may be a good improvision, but again is undesirable as a routine to divert combat forces to such tasks.

NON-AVAILABILITY OF ADVANCED AIRFIELDS, WHAT THEN?

In a nuclear age, advanced air bases to act as a link in this transportation scheme may not be available. To overcome this, a seaplane capable of flying from CONUS to distant task forces and landing in the open seas should be considered. With a hatch opening in the top of the fuselage, helicopters perhaps could hover and lift the air freight and transfer it to a mobile logistic support ship for sorting and further delivery to the requesting unit. Again the CVUK fits into the picture. The Martin Sea Master appears to offer food for thought in this regard.

In considering such a high performance seaplane, the thought may arise that this is a big plane to haul around small packages. There is no way to get away from it, the emergency requirements in the Fleet today generally come in small packages. On the other hand the ruggedness necessary for open sea landings and take-offs has not yet been built in any-

thing smaller. We must then pay a high monetary price for this delivery capability to fulfill a high military worth to combat forces. This is really no different from the way other units operate, when at times they provide a little or no direct return—patrol planes on routine flights with nothing to report, destroyers steaming for days filling a slot in the screen, or ASW forces looking, but not finding an enemy. Military worth just cannot be measured in dollars.

This high dollar cost to deliver small packages can be moderated to some extent, however. The seaplane could be very useful for delivery of high priority mail and personnel and for evacuation purposes.

A one man, portable helicopter, capable of being air-launched, carrying a one man crew and a little baggage, and being able to return to a flying aircraft, may offer some possibilities for the last linkup between the pipeline plane and the CVUK. This would preclude the need for open sea landings in many cases and thus may modify the details of the above discussion somewhat.

SENSITIVENESS OF PIPELINE AIRCRAFT TO COMBAT COMMANDER IS PARAMOUNT

For acceptable effectiveness it is considered essential that the pipeline aircraft be extremely sensitive to the requirements of the task force. For this reason centralization of the control of the aircraft must be no higher than the level of the Commander Mobile Logistic Force. To fit this scheme into the normal overseas airlift procedures, FLOGWING or MATS freight terminals, and the insistence on full payloads will defeat the entire purpose. You may be able to justify full payloads and higher centralization of control from a monetary standpoint, but hardly from a military worth standpoint in this instance.

Here also the closest tie-in between requisition control and movement control is essential. Any breakdown in the rapid flow of information between the activity overseeing the processing of the requisition and the activity planning and controlling its movement to the ultimate consumer could only result in ineffectiveness. This leads to the conclusion that requisition control and movement control must be performed by the same activity, which in turn is responsible to the officer charged with providing the logistic support to the operational commander.

It is not beyond my thinking that an officer would be attached to the seaplane who would be the requisition and movement control activity, personally shepherding the requisition from the moment he receives it until he witnesses its delivery to the CVUK. This would reduce to a minimum the possibility of an administrative breakdown in the handling of the necessary papers and the material.

As thoughts in Naval circles turn to nuclear powered, single ship "task forces" or two or three ship task forces, ranging far from conventional support, the utilization of aircraft for more Fleet support purposes, in my opinion, cannot be ignored. High speed replenishment ships of the one stop concept (less than bunker fuel) may be the best concept to replenish the bulk, but a highly sensitive, fast and long range support system would become more and more a necessity to support a ship or even small groups of ships in such advanced states of technology.

EQUIPMENT, BATTLE READY, OF HIGHEST IMPORTANCE

As our Navy grows more and more dependent upon precision made equipment to wage and win battles, the lack of the equipment in working condition becomes more and more serious. Thus the effectiveness of fleet logistic support must continuously approach perfection. As time and space can be tools for the combatant commander, it is axiomatic that they also be made to work in favor of logistic support effort.

Can we afford such a scheme? A more pertinent question is, can we not afford it? "For the want of a nail, the shoe was lost; for the want of the shoe, the horse was lost;—for the want of the battle, a kingdom was lost; and all for the want of a nail."

AN INTERPRETATION OF RESEARCH AND EXPERIENCE IN STRUCTURAL WELDING

ACKNOWLEDGEMENT

This is a slightly abridged version of the 1955 Adams Lecture given by La Motte Grover before the Fall meeting of the American Welding Society in Philadelphia on October 17, 1955. The lecture dealt principally with the brittle fracture problem. Reprinted from the March, 1956, edition of "The Welding Journal."

THE THEME for this lecture is a discussion of some of the things we have learned from research and experience in matters related to welded steel structures, and how it seems to me they can be applied appropriately to the solution of problems that arise in designing and building welded structures.

Structural welding has had its growing pains, just as other methods of construction have had theirs. Difficulties with any method may arise and prevail for a time. Sometimes they recur when changes are made in materials or methods—for example, the unforeseen reactions that still occur now and then between certain aggregates and portland cement materials and result in permanent expansion or growth of concrete.

In such cases, the engineering profession becomes apprehensive and doubtful for a while. Then, from all outward appearances, the troubles are cleared up rather suddenly in an apparently simple manner. However simple the methods of correction may seem, their discovery through research and study has usually been anything but simple. They may involve details of design and construction practices, but very often it is the improvement and better control over materials that proves to be the major factor.

Apart from non-structural factors such as corrosion, there are five rather well recognized possible causes of failure, serious structural damage or limited usefulness of engineering structures such as steel bridges, ships, building frames, railway car underframes, heavy construction equipment, and similar fabricated steel construction:

1. Instability or buckling of compression members or elements.
2. Excessive elastic deflections.
3. Fatigue (stress repeated many times).
4. Plastic deformations of such a character or degree as to result in a serious impairment of the usefulness of a structure.
5. Brittle fracture.

Contrary to conventional design assumptions, the attainment of a localized yield point stress does not, in itself, constitute a significant impairment of the strength of a structure. In any practical engineering structure there are inevitably a good many points of comparatively high concentrations of stress; and localized plastic strains occur under the first applications of normal design loading. They are the result of the geometric shape of almost any kind of useful members and their connections, and of unavoidable imperfections in fabrication and erection, and in conventional methods of design. It is important to differentiate between these localized yield point stresses and comparatively widespread field of yield point stress.

These localized concentrations of stress and strain are not recognized in conventional analysis and design by the elastic theory. Even in the newer plastic theory of analysis and balanced design of continuous frame structures, and in the evaluation of the effect of secondary stresses, it is only the less highly localized plastic strains at certain locations that are recognized.

Research at Lehigh University into the plastic behavior of large structural steel members and rigid connections has contributed a great deal of valuable information toward the practical use of the plastic theory of design. Their tests have included tests to destruction of corner connections or "knees" involving sections as large as 36 inch deep, wide-flange beams, as well as tests of complete rigid frame bents.[1]

The limitations of this method of design have been investigated. The possibility of fracture has been studied, as well as instability or buckling of compression members or elements, and excessive deflections or deformations. Rules of design and application have been formulated to assure plastic behavior, as as-

sumed, and to preclude structural damage or excessive deformations or deflections.

In the advanced stages of loading of continuous frames designed by this method, plastic "hinges" occur, successively, at the various locations of maximum bending stress, thus redistributing bending moments to other less highly stressed portions of the structure. The result of applying the method is a balanced design in which each member or structural part has, as nearly as possible, the same useful strength as any other member or part—that is, the maximum useful moment capacity of each member is utilized. The material is therefore used efficiently. The resulting design is based upon the true ability to carry load. Only by the use of welding can the economic benefits of this method be fully realized.

Thousands of tons of steel are wasted every year in designing steel building frames by the conventional elastic theory, because it results in some members or their parts being over designed, as compared with other parts of the same structure, with respect to their true ultimate carrying capacity.

Studies and test results indicate that the strains accompanying the formation of the plastic hinges, in usual cases, do not exceed 1.5%; so no appreciable strain hardening or serious loss of ductility would be expected, unless there are severe notch-like-details or defects in the affected area.

In general, all of the practical engineering methods of design have been quite satisfactory for providing safe structures, when applied in a rational manner based upon a good understanding of the behavior of materials. The structural steel material used in such structures usually has retained enough true ductility or notch toughness, after undergoing the various operations of fabrication and erection, to accommodate itself to the actual conditions imposed in service.

In various research and study projects, the Structural Steel Committee of the Welding Research Council and its Fatigue Testing Committee have investigated a number of the factors that have been mentioned as possible causes for serious structural damage. Their tests have all been of specimens that simulate actual structural members or their parts. These specimens have been simple from an engineering viewpoint, although complex from the viewpoint of the scientist who is studying fundamentals.

The objectives have been to develop the kind of engineering information that can be applied in practical design and control of materials and construction, and in writing some of the codes, specifications, manuals and handbooks covering recommended practices in structural welding.

The results of some of these testing programs afford a good deal of information that can be used for correlation with fundamental information developed through the use of less complex specimens, for example, in studying the problem of brittle fracture.

They are especially useful in this respect because there have been comparatively few service fractures or serious difficulties in actual welded structures such as bridges, constructed of American structural steel. Consequently, there is a dearth of information for use in correlation, based upon adverse service conditions.

THE BRITTLE FRACTURE PROBLEM

The discussion of brittle fracture in this paper will be limited for the most part to an attempt to correlate research findings with experience, in such a manner as to be of special interest to bridge and structural engineers.

Most of the intensive studies of the brittle fracture problem, made since the early part of 1943, have been associated largely with investigations of ship failures or the behavior of military weldments, and more recently the behavior of pressure vessels and tanks. However, the problem is an old one. Most of its general fundamental aspects have been recognized for many years, even before bridge engineers in Germany and Belgium encountered similar difficulties in 1937 and 1938.

In a critical survey by Shank of brittle fracture in carbon steel structures other than ships,[2] information was foundd on such failures that occurred as long ago as 1879. One of his most interesting conclusions was that "there is no evidence available to demonstrate that the percentage incidence of brittle failure in non-ship structures has either increased or decreased with the advent of welding."

The general basic factors involved in this problem have long been recognized as:

1. Notch ductility or toughness characteristics of the materials.
2. Service temperatures.
3. Localized state of stress as influenced by geometric shape and proportions, as well as loading.
4. Localized rates of application of stress.

By taking extreme precautions in connection with only one of these factors, an engineer might preclude brittle fracture, but this would not be a practical solution. For example, one might specify a special low carbon, quenched and tempered alloy steel, or some other kind of special heat-treated steel with very exceptional fracture propagation resistance, for general use in bridges and other structures exposed to comparatively low winter temperatures. Apart from the prohibitive cost, the steel mill capacities for producing such material would fall very short of meeting the demands.

One might go to such extremes in design and methods of construction as to reduce notch effects to the very minimum. This would not only be prohibitive in cost, but in many cases it would impair seriously the usefulness of a structure because of unfavorable arrangement.

The practical engineer is continually confronted with the problem of how to write his specifications to provide an adequate margin of safety at a reasonable cost. There is a calculated risk involved in the proper solution of any engineering problem, but it is important that it be a calculated one. Research in

welded structures has done much to provide for rational methods of calculating the risk of brittle fracture.

A great deal has been learned from fundamental research during the last decade about the metallurgy and mechanics of brittle failure. Much of it can be applied in choosing materials and in designing practical structures to avoid such failures. As in other engineering problems there must be a compromise between idealism and practicability. It is here that the scientists and engineers sometimes find it difficult to agree.

Brittle failures in actual structures have yielded some valuable information, at least in the form of statistics that have been useful for correlation with laboratory test data. Complexity and lack of complete information usually make it impossible to derive fundamental information from such failures. Also, the predominance of one influential factor may overshadow others to the point that little information of educational value can be derived.

The steel material involved in the early welded bridge failures in Europe was generally of such poor notch ductility and uniformity, that little fundamental information was gained from these failures at the time. Even then, the "red herring" of localized residual stresses due to welding was injected so strongly into technical discussions that, to say the least, it is still accorded much more recognition in some circles than it seems to merit. Many attempts have been made to demonstrate by laboratory research that residual welding stresses have been an important factor in brittle fractures. Attempts have been made to establish a correlation between such residual stresses and failures of actual structures. Certainly, none of these attempts has been successful in developing conclusive evidence.

There is some evidence that widespread areas of residual stress, called "reaction stresses," may contribute to the propagation of a brittle fracture, once it has been initiated. Also, it has been thought that they might exert an influence when very high rates of stressing are involved. However, it is well known that such comparatively wide fields of residual stress have always been present in steel structures. They result from non-uniform cooling of shapes and plates in the steel mill. When proper welding sequences are followed, the residual stresses resulting from the welding are localized rather than wide-spread. Experience has shown that very high secondary stresses, for example those in heavy truss members at riveted connections, can be neglected in design. Like residual welding stresses, they are self-limiting. As major emphasis is coming to be placed upon preventing *initiation* of fracture, the practical importance of residual welding stresses would seem to disappear.

It is important not only to differentiate between localized stresses and wide-spread fields of stress, but also to recognize the difference between temporary shrinkage or temperature stresses and so-called residual welding stresses. The former may occur at a time when the strength and ductility of the heated metal is greatly impaired. Their pattern may be different from that of the residual stress after the weldment has cooled to a uniform temperature.

It is true that fractures have started in welded subassemblies that are subjected to little if any, external loading, some little while after welding operations have been completed or suspended. However, these instances usually occur under changing conditions of ambient temperature, and frequently when localized heat is introduced with the restarting of the welding in an incompleted joint. Conditions of temporary support or restraint of a weldment, as well as influences from temperature, may result in stresses that provide an important contribution to the initiation of fracture.

However, if there are shrinkage cracks or other deficiencies in a weldment, which constitute such severe conditions that they can start a fracture under the conditions mentioned, then they would likely result in later fracture anyway, under normal service loads, apart from any influence of residual stresses.

If a member or connection, by reason of faulty details of design or defects of workmanship, cannot sustain the slight amount of straining required for mechanical relief of localized residual welding stresses, then it could hardly be expected to sustain the small localized plastic strains that inevitably occur in any practical engineering structure.

TRANSITION TEMPERATURES

It was not until the late 1940's[3] that it became well recognized that steel can be considered to have two types of transition temperatures, with respect to brittle fracture: (1) a ductility transition pertaining to initiation of brittle fracture and (2) a fracture transition temperature pertaining to propagation of fracture.

The ductility transition, related to *initiation* of fracture, is of course dependent upon the sharpness of notch-like defects. It must be dependent also upon the localized rate of increase in stress, which exerts a pronounced effect upon the yield stress.

The velocity of impact loading or rate of applying stress is much higher in some structures than in others. Impact tests of beams at Columbia University, which will be discussed further on, have demonstrated that under impact, higher yield stresses are developed. It is all right to exaggerate testing conditions such as impact loading, and to include the effects of the sharpest possible notches, to facilitate the study of a problem; but the results must not then be applied *directly* to practical design and construction, unless equally severe conditions are actually involved.

Further, experience indicates that notch acuity can be controlled in practical fabrication, and potential crack starters of maximum possible sharpness are not invariably present in welded structures. These factors must be considered carefully in adopting a ductility transition temperature that is intended to indi-

cate the lowest safe operating temperature for a structure.

Simply for the purpose of establishing a basis for reference and correlation, it is well to define a *particular* ductility transition temperature. Such a ductility transition, the nil ductility transition (NDT), has been defined (perhaps rather loosely) by researchers, along with a fracture transition, elastic (FTE). These may be characterized as follows:

1. The nil ductility transition (NDT), associated particularly with the *initiation* of brittle fracture (see Fig. 1). It is the temperature below which the behavior of a particular steel material *under impact loading* becomes so brittle *in the presence of the sharpest possible notch* (a cleavage crack notch) that fracture will be initiated with no significant prior plastic deformation at the root of the notch. This NDT is some 50° F below the FTE, when these transitions are determined by explosion tests and drop weight tests such as those conducted at the Naval Research Laboratory.

2. The fracture transition, elastic (FTE). It is the highest temperature at which brittle fracture, once initiated, will propagate through an elastically loaded region. Propagation may occur in a field of comparatively low stress, perhaps as low as 5000 psi in the case of ordinary structural steel.

For less acute notches, such as imperfections in well constructed bridges, the ductility transition seems to occur at a considerably lower temperature than that determined by laboratory tests that have been used to determine the above-mentioned NDT. Further, experience as well as impact tests and static tests of beams indicate that when the rate of increase in stress is decreased by decreasing the velocity of loading, the ductility transition is lowered a good deal. The rate of stressing in most bridge members is not high as compared with that of the impact tests of beams at Columbia University.

The studies of Robertson in England[4] and those of Pellini at the Naval Research Laboratory in the U.S.A.,[5] have clarified our understanding of initiation and propagation factors. Pellini and his associates have demonstrated a third transition, the fracture transition, plastic (FTP), which rounds out the picture but is of little interest to bridge or structural engineers. Above this FTP temperature (which is some 120° F above the NDT) rupture of the steel can occur only by 100% shear tearing under extreme plastic loading.

The investigations and interpretations made at the Naval Research Laboratory are especially helpful to practical engineers. This research has involved specimens that more nearly simulate actual conditions in an engineering structure, but the specimens have been kept simple enough to control variables quite well. However, the rate of loading and stressing has been comparatively high.

The Charpy V-notch impact test has become widely accepted as the most practical method of determining the notch ductility of structural steel materials. Quite a good correlation has been established between Charpy-V transition temperatures and the behavior of ships in service, as well as with large scale tests of the more commonly used structural steel materials, such as wide plate tests and the NRL explosion bulge tests and drop weight tests.

The proper Charpy-V energy level to be used as basis for determining the fracture initiation temperature for any class of structures in service varies considerably, depending upon service conditions such as expected rate of stress application, that is, how much impact is involved in the loading, as well as the severity of notch-like defects. It also depends upon deoxidation practices and other characteristics of the steel making process. For structural carbon steels, energy values from 5 to 20 ft-lb have been proposed by researchers, the higher value for fully-killed steel (see Fig. 1). Correlation with tests of large specimens has indicated that the appropriate Charpy-V energy values for some of the special heat-treated alloy steels of exceptionally good toughness are considerably higher than for structural carbon steel.

In view of the excellent service records of a good many welded bridges in cold climates, these proposed values seem to peg the lowest safe operating temperature at an unduly conservative figure for steel bridges, welded or riveted, for which reasonable care and control are exercised in design and construction.

THE ROLE OF WELDING IN BRITTLE FRACTURE

At the Naval Research Laboratory, Pellini and Eschbacher have evaluated the notch ductibility characteristics of various common types of weld metals.[6] Explosion bulge tests have shown that it is only when the notch toughness of the base metal is greatly superior to that of the weld metal, that fractures can be made to follow welded joints (unless the temperature of testing is quite low).

By means of drop weight tests, the E6010 types of welds were found to be somewhat superior to the semi-killed and rimmed structural steel plate of ¾ to 1 in. thickness used in these tests, and essentially equivalent to the as-rolled fully-killed steel plate. The E6102 type appeared to be at least somewhat inferior.

The E7016 type, as well as the E6020 type, was found to be essentially equivalent to the E6010 type. This seems to indicate that the superior behavior of members welded with E7016 low-hydrogen electrodes is derived largely, if not entirely, from a

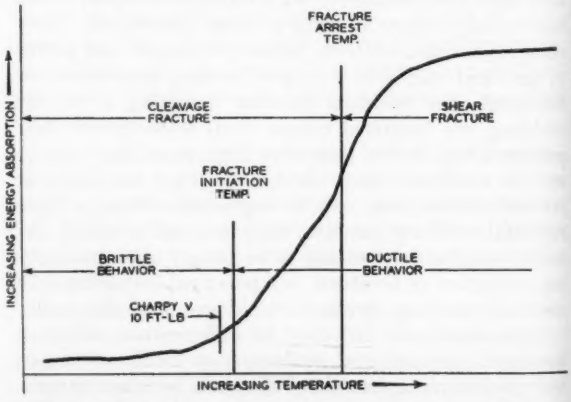


Figure 1. Behavior of structural steel under impact loading in presence of sharpest possible notch.

diminution of the severity of flaws, rather than from improved metallurgical conditions.

Although the metallurgical effect upon base metal of the heating and cooling cycle during welding may be important with respect to the formation of shrinkage crack flaws, even for ordinary carbon structural steel, perhaps a disproportionate amount of importance was attributed to this factor in the earlier studies of the brittle fracture problem.

In interpreting some of his later studies, Pellini states: "The basic role of welding in the development of brittle fractures is now clear. In or near the welds, crack-like flaws may be developed for a variety of reasons. When the tips of such notches extend into the base steel, the notch ductility of the steel becomes the governing factor."

There is some evidence from practical experience to confirm this belief. A thorough examination of sources of fractures in welded structures has sometimes disclosed unmistakable evidence of sizable crack-like flaws at welded joints. Further, it has been found that service fractures correlate with the transition temperature of the unaffected base metal, rather than with the characteristics of the weld or the adjacent heat-affected zone.

SERVICE FRACTURES IN WELDED AND RIVETED STRUCTURES

Research has provided information that helps one to interpret better the behavior of steel that has been observed in engineering structures that have failed.

Brittle fractures are found almost invariably in the heavy material of any large steel structure that collapses. They may be found, even in the comparatively light material of component parts of large riveted members, and when the prevailing temperature during collapse is not low. There has been a tendency among engineers to consider the brittle granular appearance of such fractures to be, in itself, an indication of inferior structural steel material. However, under such conditions of collapse, the tremendous over stress would be expected to force initiation and propogation of some fractures of quite brittle appearance, even at temperatures 70° F or more above the lowest service temperature that would be safe for ordinary conditions of load and stress.

In January 1951, about four years after the Three Rivers Bridge had been put in service in the Province of Quebec, Canada, a local shattering, of an appearance such as would be expected from a demolition bomb, precipitated the collapse of several continuous spans of this large welded girder bridge when the temperature was 30° F below zero.[7] The local shattering simply started a mechanical "chain reaction" that caused the successive collapse of the spans of one continuous unit, letting them drop, one by one, to the river bed. The steel girders were broken in many pieces with fractures of the brittle granular type, as would be expected. However, even at this low temperature, many parts of these heavy girders were bent most severely without fracture. In evaluating the significance of the failure of this bridge, it is important to note some very unusual features and conditions, apart from the fragmentation in one localized area which convinced demolition experts that the failure was due to sabotage.

Evidence was developed by disinterested parties, leading to the belief that the concrete of the bridge floor was of the expansive type mentioned in the first part of this paper, which exhibits continued permanent growth in volume. The concrete floor was quite well anchored to the girders by shear developers and floor expansion devices, and also by the embedment of the top flanges of the floor beams.

A structural analysis made by Lamb[7] has shown that as a result of such growth of concrete, widespread reaction stresses of greater than normal yield point intensity might be created in steel girders of the proportions involved in this structure.[7] Under the prevailing circumstances, such stresses could initiate fracture and would surely be an important contributing factor.

In a previous investigation, prompted by the occurrence of some local fractures in the girders of this structure, about a year prior to its failure, some of the steel material at highly stressed locations, including those of initiation of these previous fractures, was found to be a rimmed steel with pronounced segregations. Although it conformed essentially to the provisions of the ASTM-A7 specification, which has been used commonly for bridge construction in the United States and Canada, this steel was of quite inferior quality as compared with that normally supplied under this specification. Further, the carbon content of some of the other steel approached 0.40%, as determined from drillings taken from mid thickness. Some of the subsequently adopted ASTM specifications for structural steel provide definite requirements for carbon and manganese. The ASTM-A7 specification does not. The characteristics of this steel material would be expected to raise the temperature for initiation of fracture a good deal above that for a normal welded bridge.

One observation of the general behavior of this structure during collapse is of interest in connection with paths of propagation of brittle fractures in welded structures. A thorough examination of the wreckage failed to reveal any tendency of the fractures to follow any of the welds or their fusion zones or heat-affected zones, despite the low temperature of—30° F. This behavior is quite typical of the collapse of structures with reasonably sound welds, although very severe over stresses must exist in almost every conceivable direction during collapse.

Experience of this kind in ships as well as bridges has led to the belief that it has been the notch ductility of the base metal, rather than that of welded joints, that has constituted the controlling factor with regard to brittle fracture. Such experience confirms the findings from research at the Naval Research Laboratory that have been mentioned in connection with the role of welding in brittle fracture.

These research findings have been confirmed also by the previous investigation made when two frac-

tures were discovered in 2-in. and 2½-in. thick flanges of the girders of this same bridge, about a year prior to the failure. Traces of shop paint were found in both of these fractures, indicating the presence of small crack-like flaws that extended slightly into the base metal.[8]

The unusual characteristics of the steel material of this bridge would increase the susceptibility to the formation of such crack-like flaws. A current revision of the American Welding Society Standard Specifications for Welded Highway and Railway Bridges will not only specify the use of one of the newer structural steel materials, ASTM-A373, but it will also include more definite provision for welding procedures.

Soon after completion of the Zoo bridge in Berlin in 1935, when the temperature was 14° F, several transverse fissures were discovered in the complete penetration web-to-flange groove weld T-joints of the girders. A laboratory examination lead to the belief that the fissures had started from shrinkage cracks that occurred during shop fabrication.

Before the Rüdersdorf bridge in Germany had been opened to traffic, in January 1938, two serious brittle fractures of the girder webs started at similar T-joints adjacent to tension flanges, at a temperature of 10° F. It is well known that the structural steel material used in general for bridge construction in Germany and Belgium at that time was of much poorer weldability and notch-toughness than the American ASTM-A7 steel, and even more inferior as compared with the quality that is assured by the newer American structural steels for welding; AS TM-A131 and ASTM-A373.

When static load tests were made of welded girders, built of the same kind of steel material as was used in the early welded German bridges, to investigate the causes of fracture in these structures,[9] the presence of very small, virtually microscopic crack-like flaws did not prevent a large amount of plastic deformation without fracture at testing temperatures carried as low as —22° F.

A comparison of the report on these girder tests with descriptions of the service fractures in the literature,[10] seems to indicate one important difference. The shrinkage cracks found in the Zoo bridge, for example, were quite sizable ones extending slightly into the base metal. The crack-like flaws that did not cause fracture in the test girders were virtually of microscopic proportions, or at the most of very small size, say 1 mm in length.

Such test results no doubt explain the reluctance that has prevailed, to believe that so-called inherent flaws of almost microscopic proportions are important factors in the initiation of brittle fracture in ordinary engineering structures.

The crack-like flaws discovered in the Three Rivers bridge in Canada were small flaws, but not of the extremely small or microscopic kind. The difference is important, because microscopic crack-like flaws are likely to be found in any cast metal such as a weld.

Pellini is emphatic in differentiating between obvious "chance flaws" in welds, and the "microscopic flaws inherent to cast weld metal." His explosion tests showed that in the presence of a sharp notch (presumably comparable to a chance flaw), any inherent microscopic flaws exert a negligible or very small influence upon the temperature at which fracture will initiate with no measurable deformation, although they do exert some influence when there are no larger chance flaws to overshadow their influence.

Thus research as well as experience seems to indicate that the practical bridge or structural engineer need not be concerned about microscopic flaws in welds or base metal. More important flaws or imperfections are inevitably introduced by any of the various fabricating operations such as shearing, punching, bending, flanging, friction sawing, rough machining, welding and rough manual oxygen cutting.

All of the fractures of welded girders that have been mentioned seem to have started at the welded T-joint between the web and one of the flanges. Some of these T-joints have been fillet welded, and others have been designed as full penetration groove welds. Such joints surely constitute critical locations. This may be caused, at least partly by restraint and multiple paths for conducting heat away during cooling after welding, which may result in crack-like flaws when welding procedures are not properly controlled. However, it should be emphasized again that the crack-like flaws or small fissures that have been discovered in the investigations of fractures in bridge girders have been obvious, though small, rather than microscopic flaws. Experience has indicated that it is at least somewhat more difficult to avoid crack-like flaws in making a complete penetration T-joint than in making a fillet welded T-joint.

Apart from the influence of any crack-like flaws that may exist in the web-to-flange joints of a beam or girder, rather severe concentrations of stress and strain have been found at such joints. More research into this behavior at T-joints would be desirable.

EFFECT OF DETAILS OF DESIGN UPON DUCTILITY

There is a sufficient background of service experience to show that good cold weather performance can be expected of welded bridge girders, when they are properly designed, of suitable inexpensive materials, and welded with properly controlled practical procedures. The details of design need not be arranged meticulously nor refinements carried to the extent of impracticability, to accomplish this. However, the designer must be on the alert to avoid extreme degrees of constraint and concentration of stress.

For example, the structural arrangement for a highway grade separation structure built a number of years ago, involved a number of intersections of main longitudinal girders with transverse pier girders, both of which carry quite large negative bending

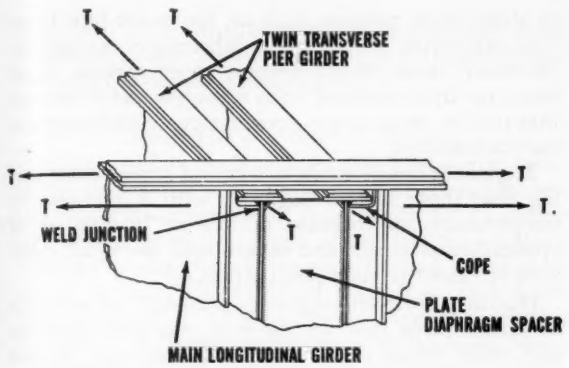


Figure 2. Intersection of longitudinal girder with pier girders of highway overpass structure.

moments at the intersection near a point of support on a pier. To avoid a field of biaxial tension in the plane of the upper flanges, the web of the longitudinal girder was coped rather generously (see Fig. 2) to provide for the upper flanges of the twin transverse girder to extend through this cope, just below the flange of the longitudinal girder, thus avoiding a direct intersection of the flanges.

The twin transverse girder-members are tied together at this point by means of a diaphragm in the plane of the web of the longitudinal girder, making this web continuous. Thus, in effect, the webs of the girders do intersect.

Therefore, the upper part of the diaphragm, is in a position to be stressed in tension due to the negative bending moment of the longitudinal girder. At the point of intersection of the girders, the upper parts of the webs of the twin transverse girder are likewise in tension due to negative bending moment. The line of intersection of the webs in this region is, therefore, subjected to quite large tension stresses applied in the two directions.

If the diaphragm spacer of the twin girder and its welded connections had been detailed to provide a slight amount of flexibility, this would have relieved the concentration of biaxial stress and the high degree of local restraint at the upper part of the web-intersection.

The unfavorable condition was further aggravated by the cope in the web of the longitudinal girder at this point being made hardly deep enough to clear the flange-to-web fillet weld of the transverse girder. This resulted in a total of eight lines of fillet welds coming together almost at one point, in a field of biaxial tension due to load. In the returning of the vertical fillet welds across the thickness of the coped web, these returns caused a marked degree of almost unavoidable undercutting in the horizontal fillet welds at this same point, thus creating at least a dull notch effect.

Experience has shown that fractures start sometimes from points where several welds intersect as in the detail that has just been described. It may be that such an arrangement makes it difficult to avoid crack-like flaws due to unfavorable welding conditions together with multiaxial restraint against shrinkage while the welds are cooling.

After the bridge had been in service some twelve years, a fracture was discovered during a routine maintenance inspection. It started at the above described weld junction, at a web-to-flange joint in one of the twin transverse girder-members, apparently at the undercut. The fracture extended downward in the web to about midheight. A close examination showed that it had also penetrated slightly into the under surface of the 1⅜ in. thick flange plate.

A careful inspection of other similar details on the same bridge revealed no other flaws or fractures. Even this unfavorable detail of design apparently constituted a marginal case for the temperatures involved, in which the probability of fracture was not very great. During the twelve years of service before the fracture and two subsequent years, sub-zero temperatures must have been encountered quite a number of times.

Another somewhat similar case has been reported in which a crack started at a trihedral corner where a load bearing stiffener at a continuous girder support was welded to the upper tension flange of the girder, as well as to the web. Since the inner corners of the stiffeners were not clipped, this resulted in the same kind of intersection of fillet welds as has been described.

In the light of experience and research findings, it should not be too difficult to select details of design that will avoid excessive local concentrations of biaxial stress and restraint.

EFFECT OF FABRICATION OPERATIONS

That serious impairment of notch ductility due to fabrication is not associated particularly with welded structures is evident from failures that have occurred in riveted ships and other structures.

World-wide data on ship failures reported recently in a London publication,[11] showed that between 1947 and 1952, fifteen ships of between 2,000 and 10,000 gross tons, either broke in two or foundered, following major structural failures. Four of these were riveted. Most of the other eleven, which were welded, had been built under unusual wartime conditions.

Not long ago serious fractures developed in both of two 200-foot truss spans of a riveted highway bridge in one of our northern states. The bridge was erected during the winter of 1949-1950, at a location where the temperature dropped as low as —43° F, while the bridge was in service about two years, prior to the discovery of the most serious fractures in the spring of 1952.

All of the fractures started at the rough cold worked edges of raw punched rivet holes, evidently after fabrication had been completed, because there were no traces of shop paint on the fracture surfaces. Check tests of mechanical properties, together with reports on chemistry, as well as metallographic and arc spectrum examinations, failed to disclose any abnormality of the A-7 steel material.

These most serious fractures, which occurred in service, were near midspan in a bottom chord tension member, composed of two 15-in., 40-lb. channels. This thickness of material in these channels is only 9/16 to ⅝ in. Brittle fractures in service seldom occur in members of such proportions and thickness. Both channels were completely fractured, with breaks quite straight across. In one channel the fracture was through rivet holes at both the top and bottom of the channel.

A fracture at midlength of an upper compression chord near the middle of the other span had been discovered previously during painting, following erection and removal of falsework, but before the concrete floor had been placed. This fracture started at a bottom flange tie-bar rivet hole and extended upward to the top flange fillet. The fracture was on the concave side of a slight lateral bow in the member. The location of the fracture suggested that it might have started during handling and erection, without detection, and the member may have been overstraightened subsequently. The temperature was probably not lower than —18° F at the time of this fracture. Handling during erection could well have produced sufficient localized strains at the edge of the rivet hole to force the initiation of fracture at the prevailing temperature.

These fractures could likely have been averted if one of the more recently available structural steels, A131 or A373, had been available and could have been used. They could almost surely have been averted if, in addition, the rivet holes had been carefully reamed or drilled, to remove the crack-like flaws and cold worked material around the periphery of the punched holes.

There may have been some other influential factor, such as a particular susceptibility of the steel material of these bridges to strain age embrittlement, which was not disclosed by the investigation. This is suggested in view of the fact that no factors were discovered that would single out this bridge as unique. There have no doubt been a good many other steel bridges fabricated with raw punched holes and built in the northern part of the United States, of the same kind of ASTM-A7 steel. However, research that will be discussed further on in this paper, has shown that shearing and punching impair notch ductility very drastically.

STATE OF STRESS

In analyzing the brittle fracture problem, the term "state of stress" has been used commonly to include only those factors relating to geometric shape and direction of application of stress. These factors determine whether the stress is uniaxial or multiaxial and whether the total stress is quite severely concentrated at some points rather than being more widely distributed as assumed in conventional design.

For practical purposes of study and analysis, it has been convenient to investigate separately the effects of sharp local notches such as the crack-like flaws that can result from various fabrication operations. However, these effects become more serious when they are super-imposed upon other geometric factors that tend to concentrate stress seriously and suppress ductile behavior.

Engineers like to think in terms of stress, considering that these factors, together with a decrease in temperature, an increase in rate of loading or of application of stress, and severe cold work, all combine to raise the yield point stress.

Ductility or notch toughness is thought of as a sort of safety valve that will permit secondary stresses and other stress concentrations to become "ironed out" before the stress rises to the point of fracture, provided that plastic deformation is not inhibited by excessive restraint or triaxiality, low temperature or high rates of stress application.

It is known that fractures can be started at temperatures quite far above the ductility transition temperature of a steel material, if the stress and the available energy are sufficiently high to force initiation of fracture, after relatively great local plastic deformation at the root of a notch. Under these conditions a ductile tear presumably can undergo a crack sharpening process if the temperature is sufficiently low. Strain aging as well as strain hardening may be an influential factor in some cases.

On the other hand, there seems to be evidence that under some circumstances, and up to a point, plastic straining can in some manner improve the cold temperature performance of a structure, particularly if the straining is not great and it is done slowly at relatively warm temperatures.

There has been some well founded conjecture as to whether some benefit may not be derived from the proof testing of structures during relatively warm weather under a slowly applied loading substantially in excess of the design load. Such proof testing or "shake down" of both riveted and welded bridges has been quite common practice for many years in Europe. For continuous frames designed by the plastic theory, such a "shakedown" might be advisable, if they are to serve at rather low temperatures.

Experience records of welded ships do not seem to indicate any general pronounced reduction of the probability of brittle fracture with increased service life. However, it is usually difficult to determine, in the case of brittle failures in service, whether the structure has even been subjected to a prior heavier loading at a warm temperature, even though it has been in service for a number of years.

In interpreting the results of practical experience as well as research, a careful differential must be made between initiation and propagation of brittle fracture. Comparatively small tears, or even short granular cracks, may be forced to occur at notches, without starting a fracture, because the temperature, stress and elastic energy conditions do not permit its initiation. Also, short shrinkage cracks may occur in a weld, due to unsuitable welding procedures; but

this does not constitute initiation of fracture. Such shrinkage cracks are not believed to be as sharp as a cleavage crack that is propagating.

DROP HAMMER TESTE OF BEAMS

In a project of the Structural Steel Research Committee completed not long ago at Columbia University under the direction of Prof. Krefeld,[12] an extensive investigation was made of the low temperature behavior under drop hammer impact of rolled steel beams 12-in. WF at 71-lb., of 12-ft. span, with butt-welded splices at midspan. These specimens of 13/16-in. flange thickness and ½-in. web thickness were adopted as realistic prototypes of structural members. The program included the testing of some unwelded beams without splices but with semi-circular holes cut in the webs of the beams adjacent to the flanges, to simulate the geometry of most of the welded beams, in which such copes were provided to facilitate the making of sound butt welds in the flanges.

Some of the beams were of ordinary ASTM-A7 semi-killed steel (0.21% C, 0.57% Mn and McQuaid-Ehn grain size, 100% 1-3). The material of the other beams was of similar tensile properties conforming to ASTM-A7, but it was fully-killed steel with about 0.21% silicon and 0.047% aluminum, and had a somewhat higher manganese-to-carbon ratio (0.19% C, 0.72% Mn and McQuaid-Ehn grain size, 100% 6-8).

The beams were tested under the impact of a 2,000-lb. drop-hammer at various temperatures. The "transition temperatures" of Table 1 were determined upon the basis of a change in mode of failure, from buckling to fracture, except for a few beams, most of which were unwelded beams with a machined cope in the web adjacent to the tension flange, and with a sharp pressed notch in the top edge of the cope. For these beams the "transition temperature" was considered to be the lowest temperature at which failure would occur by buckling, before small tears or cracks, occurring at the root of the notch, would become extended to quite an appreciable length of granular fracture (greater than about one inch).

In many beams, local strains were determined, as close as possible to the origin of fracture in the vicinity of the cope near the bottom flange. As the temperature of testing was decreased, there was a marked decrease of the plastic deformation preceding fracture, in both the webs and the flanges as was expected. The deformation in the flange varied from about 0.2% at —115° F to 8% at + 30°, with a tendency for strains at the edge of the flanges to be smaller than those closer to the junction with the web, particularly at lower temperatures. Strains in the web, directly over the hole and close to it, varied from 0.2% at —115° F to 26% at + 30° F.

These data showed a concentration of stress and deformation in the web-to-flange region, with particularly high concentration in the web directly over the semi-circular cope, where most of the fractures originated. This led to the testing of some rolled beams with mid-span splices made by a more costly, but still

TABLE 1—*Transition Temperatures* (in Degrees Fahrenheit) of Beams Tested under Drop-Hammer Impact*

Beam condition	*Fully-killed steel beams*	*Semi-killed steel beams*
1. Welded with E6011 and E6020 electrodes:		
(A) as-welded	—29	— 5
(B) Welded with 350° F preheat	—57	—17
(C) Welded and Stress re-relieved	—45	— 5
2. Welded with E6016 (low-hydrogen) electrodes:		
(A) As-welded	—65	—25†
(B) Welded with 350° F preheat	—55	—36†
3. Coped web holes eliminated, and welded with E6011 and E6020 electrodes	—70	
4. Plain, unwelded beams with machined coped holes and sharp pressed notch	+25‡	+85‡
5. Plain, unwelded beams with coped web holes made by flame-cutting	—30	—60 to —115§

*Based on the temperature at which the mode of failure changes from buckling to fracture, except as noted for beams with pressed notches.
† This temperature or lower.
‡ Approximately the lowest temperature at which failure will occur by buckling rather than by propagation of a small tear or crack.
§ Too few tests to give more than a range of values.

practicable procedure, which permitted the semi-circular copes to be eliminated. This expedient lowered the transition temperature about 40 degrees. However, the same benefit might not be realized from eliminating the cope at a splice in a built-up welded girder, at a point of a high moment. The longitudinal web-to-flange welds would participate in rather high flexural stress, as well as transferring stress by longitudinal shear. It might be desirable to avoid the intersection with the splice welds.

Except for very low transition temperatures, such as the —70° F determined for beams welded without coped holes in the web, quite appreciable plastic deformation preceded the initiation of brittle fractures. Therefore one must consider the initiation to have been forced in most of the beams by a large over stress due to impact, at temperatures considerably above the ductility transition. The large over stress is confirmed by stresses that were computed by the flexure formula, from equivalent static loads that were based upon the elastic component of the observed dynamic deflection.

For the semi-killed steel, as rolled, the Charpy V-notch temperatures for a 15 ft-lb energy level were + 33° F for material in the web of a beam near the flange-to-web fillet, and + 25° F at the center of the flange. The corresponding temperatures for the fully-killed steel were—42° and 45° F. These transition temperatures varied much less for various locations in the beam, in the case of the fully-killed beams than for the semi-killed beams.

In comparing these Charpy temperatures with the behavior of the welded test beams as shown in Table 1, one must keep in mind that for the test beams, fracture initiation was forced by quite a large overstress from the impact loading; also that a good deal of deformation preceded fracture in most of the beams, at the temperatures that have been called their transition temperatures. Therefore, under ordinary service conditions, the temperature for initiation of fracture ought to be quite a good deal lower than the tabulated transition temperatures, for structures with no more serious flaws than those in the test beams.

The transition temperature for beams of fully-killed steel, as welded with E6011 and E6020 electrodes, was —29° F. For similar semi-killed beams it was —5° F. The improvement derived from the use of properly stored and conditioned low-hydrogen electrodes was very substantial and about the same as that derived from preheating (which was to a temperature higher than necessary). The two were not additive.

CORRELATION OF IMPACT TEST DATA WITH SERVICE PERFORMANCE

Since there have been very few serious fractures of steel bridges in service, constructed of material comparable to either A7, A131 or A373 steel, it is impossible to make a really satisfactory comparison of their behavior with that of laboratory test specimens. However, the similarity of the beam test specimens to actual structural members, together with the general excellent performance of many welded bridges in cold climates, may justify an attempt at correlation based upon the few failures that have occurred.

When the rather extreme forcing of initiation of fracture is taken into consideration, the results of the impact tests of beams seem to indicate that a Charpy V-notch 15 ft-lb temperature would be an unduly conservative temperature to be considered as the lowest safe service temperature for a welded structure with no sharper defect than those of these beams.

In view of the ductile behavior of some beams tested under static load at a very low temperature, as well as the results of the impact tests, and after making some allowance for the thickness of material of these beams being somewhat less than 1 in., it seems reasonable to assume that most of the tabulated beam transition temperatures for impact loading are at least 35° F above the fracture initiation temperature for normal service conditions. This would indicate a minimum safe operating temperature somewhat below —40° F for a welded structure of A7 steel when welded with ordinary mild steel electrodes, and with rather severe stress concentrators such as the coped holes, or for a riveted structure with chance flaws. This temperature is of the same order as the service temperatures at which very serious fractures occurred in two bridges that have been mentioned in this paper, one of them welded and the other riveted. Much of the steel material involved in the welded Three Rivers bridge was of abnormally poor notch ductility, and other attending circumstances that have been mentioned, were abnormal, to say the least.

The use of one of the newer structural carbon steel materials would afford an additional margin of safety. A still greater margin could be afforded by the use of low-hydrogen electrodes to assure the elimination of severe chance flaws.

EFFECT OF STRAIN RATE AND TEMPERATURE UPON YIELD STRESS

In hazarding the opinions or thoughts that have just been expressed, the writer has taken cognizance of the influence of the high rate of application of load and stress, or strain producing influence, which was involved in the drop hammer impact tests of beams. Similarly, quite high rates of application of stress must be involved in the drop weight tests and the explosion bulge tests made at the Naval Research Laboratory. Under usual service conditions of a bridge, the rate of increase of stress is comparatively low.

This effect has been termed variously as "strain rate," "rate of load application," etc. Engineers are accustomed to think in terms of stress and therefore prefer to do so.

The data taken from the Columbia University beam tests seem to indicate that one may interpret the effect of rate of load application and the effect of temperature, as well as that of notch acuity, in terms of stress; that is, if the stress or strain producing influence is exerted so rapidly that there is insufficient time for much plastic strain to develop, then the yield stress may be increased to a sufficiently high level to cause fracture with virtually no plastic deformation. It is further indicated that a decrease in temperature results in an increase in the stress that can be built up to produce fracture.

Pellini considers that the velocity or impact factor can be neglected in evaluating the results of the NRL explosion tests with brittle crack-starter welds, in view of the very severe notch effect involved. He feels that correlations he has established confirm this belief. It may be that the three factors, temperature, notch acuity and rate of stressing all contribute, up to a limit, in increasing the stress. Perhaps, in his tests, the combined effects of temperature plus the sharpest possible notch, plus a given rate of stressing may have raised the stress to a critical value that is sufficient to initiate brittle fracture with virtually no plastic deformation. In that event, a further increase in rate of stressing, could not increase the stress further because brittle failure would intervene. However, such cracklike defects of maximum possible sharpness do not seem to exist in welded construction over which reasonable control has been exercised.

It has been mentioned that yield stresses in the test beams were computed by the flexure formula, based upon the equivalent static loads that would produce the elastic components of the observed dynamic deflections of the beams.

A complete and accurate correlation of such computed yield point stresses with temperature or with rate of stress application, as influenced by the height of fall of hammer, is not possible for all of the tests. There were no means provided for measuring residual energy of the falling weight, when fracture occurred.

However, by comparing these computed yield stresses for various heights of fall and various temperatures of testing one finds quite a wide variation of yield stress from 64.8 ksi for a low height of fall of 54 in. and a high temperature of +63° F, to a value of 101.0 ksi for a great height of fall of 107 in. and a low temperature of —115° F. For tests made at the same temperature, the computed yield stress increases, in general, as the height of fall increases. For tests made with the same height of fall, the computed yield stress increases, in general, with a drop in temperature.

COMPARATIVE ACUITY OF NOTCH EFFECTS

For one series of fully-killed beams welded with E6011 and E6020 electrodes, a special welding sequence was used, which permitted the coped holes to be eliminated. These beams showed a transition temperature of —70° F. At this lower temperature of testing very little, if any, plastic deformation preceded the initiation of fracture. These beam-splice welds contained a few obvious flaws but no crack-like notch-effects.

However, when several welded beams with coped holes were tested under static loading at —100° F, they all failed by buckling, although one fully killed beam tested at this extremely low temperature developed a crack at a load just below the structural yield strength of the beam. The crack started at the bottom end of the web-splice weld, at the top of the lower coped hole. It extended to mid-height of the web, but it did not extend beyond its initial formation, nor did any fracture occur in the flange under subsequent loading to buckling. Strain measurements indicated the development of appreciable permanent deformations in the web, adjacent to the hole, prior to the occurence of the crack. Therefore even this initiation of fracture was somewhat forced.

This seems to indicate that there were no serious chance flaws in these beams, although the welding of all the beam specimens for the entire program was done under ordinary fabricating shop conditions with reasonably good but not meticulous control over the welding.

The transition temperatures as tabulated for most of the beam tests of this program might well be defined as the lowest temperature at which buckling occurred before fracture could be initiated, even by forcing at very high stress.

The beam-test transition temperatures, as specially defined for the unwelded specimens with machined coped holes and sharp pressed notches, were very much higher than for any of the welded beams. At such temperatures a good deal of local plastic deformation occurred at the notch prior to initiation of granular fractures and there were shear lips more than 0.02 in. deep. Therefore, the initiation was rather drastically forced. These particular beam transition temperatures may have been close to the FTE of the material. However, the height of fall of the hammer, required to initiate the fractures, was comparatively small. The developed yield stress may have been relatively low, likewise.

The acuity of the sharp pressed notches in these beams must surely account for initiation of fractures at much higher temperatures than for initiation in the welded beams. These notches with a root radius of 0.00075 in. could not have been as sharp as a cleavage crack notch. Further, a comparison with the results of tests of welded beams seems to indicate that none of the welds could have had flaws in them which approached the severity of the sharp pressed notch.

EFFECTS OF OXYGEN CUTTING AND SHEARING OR PUNCHING

Tests of plain unwelded fully-killed beams with coped holes made by manual oxygen cutting showed these beams to be equivalent in notch toughness to as-welded beams welded with ordinary mild steel electrodes (Table 1). For some reason, the few semi-killed, unwelded beams with oxygen cut copes had to be carried to exceptionally low temperatures to make them fail by fracture.

Tör, Ruzek and Stout have made bend tests at Lehigh University of small specimens with similar semicircular notches to study the effects of machine oxygen cutting and shearing or punching on notch toughness.[13] Transition temperatures determined on two bases, lateral contraction and energy absorption, were found to be comparable. Two steels were used, ASTM-A201 and ASTM-A285, ⅝ in. thick.

Harris has studied the same problem at the University of Illinois using static tension tests of unnotched bars ¾ in. thick, made for Committee 15, AREA.[14] His tests involved steels conforming to various ASTM specifications: A7 (semi-killed) and A7 (rimmed), A94 structural silicon steel, and A242 low-alloy high-strength steel.

The notch bend test used at Lehigh University, being a more searching and discriminating test than the tension test, showed that even machine oxygen cutting lowered notch ductility somewhat but not nearly as much as shearing or punching. It was shown that subsequent heat treatment at 1150° F improved both the flame cut and sheared edges noticeably, and heat treatment at 1600° F was especially beneficial, lowering transition temperatures for both oxygen-cut and sheared specimens to below —160° F. The authors consider this very beneficial effect of heat treatment to be strong evidence that the very adverse

effect of shearing and the moderate effect of machine oxygen cutting are not due to the formation of tiny notches at the surface, but rather are due to metallurgical changes in the metal resulting from heating or plastic deformation. However, the authors recognize the possible effect of the furnace atmosphere of the 1600° F treatment in causing scaling and sloughing off a few thousandths of an inch of metal.

Unfortunately, the program of impact tests of beams at Columbia University had to be terminated without testing any beams with oxygen cut coped holes subjected to subsequent heat treatment.

With the tension test, Harris was unable to detect any adverse effect of machine oxygen cutting on any of the steels tested other than structural silicon steel, A94. However, manual oxygen cutting "of a quality that would be expected of a novice" did lower the notch ductility of all the steels. It was believed that the quite obvious irregularities of this rough manual cutting constituted or created sizable notch effects. No attempt was made to determine whether subsequent heat treatment would improve such manually oxygen cut edges. Since it improved the sheared edges, it would be expected to improve the manually oxygen-cut edges, likewise.

The Illinois tests of all four steels showed that, by far the most serious impairment of ductility resulted from shearing. The sheared specimens of A7 semi-killed steel showed about 5% elongation in the tension test at —67° F. The impairment due to shearing of the A7 rimmed steel and the A242 and A94 steels was very great, with almost no elongation even at temperatures only slightly below +80° F. All of the sheared specimens of A7 rimmed steel fractured at very low stresses, some of them at stresses of approximately yield point value.

As in the case of the Lehigh tests, subsequent heat treatment of the sheared specimens of all steels investigated improved their ductility greatly. In fact, the University of Illinois tension tests indicated full restoration to provide strength and ductility equal to that of specimens with machined edges. Their heat treatment consisted of applying the heat from an oxyacetylene multiple flame tip directly to the face of the sheared edge. Their procedure was not controlled very accurately, but rather it was made to simulate practical shop operations. The edge was heated "uniformly and progressively to a red heat visible in ordinary shop light to a depth of at least 1/16 in." Checks with crayon-type temperature indicators which are only approximate under such circumstances, indicated that the maximum temperature achieved in this manner exceeded 1250° F.

PREVENTION OF BRITTLE FAILURE

Considerations of practicability have required a three-prong attack to prevent initiation of brittle fracture in engineering structures such as bridges, that must serve at rather cold atmospheric temperatures.

1. An improvement of steel material and its uniformity, without untenable increase in cost.

2. Selection of details of design that will avoid severe local constraint and serious concentrations of stress in the vicinity of critical points of a structure—without going to such extremes as would involve unreasonable increase in cost or impaired usefulness of a structure because of unfavorable arrangement. Experience and research both afford criteria for such a selection of details of design.

3. Control over workmanship to minimize the severity of notch-like flaws, without involving needless increase in cost due to meticulous and impracticable procedures.

The proper engineering approach for solving the brittle fracture problem seems to be indicated clearly as having for its objective the prevention of initiation of brittle fracture, rather than prevention of

TABLE 2—*Carbon, Managanese and Silicon Requirements for Plates*

	To ½ in., incl., in thickness		*Over ½ to 1 in. incl., in thickness*		*Over 1 to 2 in. incl., in thickness*		*Over 2 to 4 in. incl., in thickness*	
	Ladle	*Check*	*Ladle*	*Check*	*Ladle*	*Check*	*Ladle*	*Check*
Carbon max. percent								
ASTM—A373	0.26	0.30	0.25	0.29	0.26	0.30	0.27	0.31
ASTM—A131	..	..	0.23	0.27	0.25	0.29	0.25	0.29
Manganese, percent								
ASTM—A373	..	..	0.50 to 0.90	0.46 to 0.94	0.50 to 0.90	0.46 to 0.94	0.50 to 0.90	0.46 to 0.94
ASTM—A131	..	..	0.60 to 0.90	0.56 to 0.94	0.60 to 0.90	0.56 to 0.94	0.60 to 0.90	0.56 to 0.94
Silicon, percent (Same for A373 & A131)	..	..	...	...	0.15 to 0.30	0.13 to 0.33	0.15 to 0.30	0.13 to 0.33

Phosphorus, sulfur and copper requirements for A373 & A131

	Ladle	*Check*
Phosphorus, max. percent	0.04	0.05
Sulfur, max. percent	0.05	0.063
Copper, when copper steel is specified, min. percent	0.20	0.18

Note: for material over 1 inch thick, A131 requires fine grain practice.

propogation. Expedients like riveted crack arresters have served a most useful purpose, pending a more satisfactory solution, but they have not always accomplished their purpose.

Certainly it would be prohibitive in cost to specify steel material for the construction of ordinary bridges or ships, that would provide a fracture propagation transition temperature (FTE) comparable to the lowest anticipated service temperature. Further, experience indicates that such a provision is not warranted.

There are very few steel bridges of ASTM-A7 material that have been built in locations where the temperature never drops well below the FTE of the steel material. The vast majority serve at temperatures from 50 to 100° F below the FTE of their steel materials, many of them at temperatures far below the nil ductility temperature as determined by explosion and impact crack-starter tests. Yet brittle fracture in steel bridges, welded or riveted, has been quite rare, except when the steel material has been greatly inferior to the common run of ASTM-A7 steel, or grossly deficient welds have provided a path for a fracture to get started.

Information about propagation of fractures is of little practical importance to the bridge or structural engineer. However, it does let him know that if he does not design and build his structure in such a way as to prevent initiation of brittle fracture, such a fracture can propagate at temperatures up to the FTE (perhaps as high as 70° F) under comparatively low stress of the order of 5000 psi.

TABLE 3—*Tensile Requirements for Structural Steel for Welding ASTM Specification A 373-54T*

	Plates, shapes and bars
Tensile strength, psi	58,000 to 75,000
Yield point, min, psi	32,000
Elongation in 8 in., min, percent	21*
Elongation in 2 in., min, percent	24*

*Modified for certain thickness ranges

NOTE: Requirements for A131-53T are the same as these, except that tensile strength is 58,000 to 71,000 psi.

The performance record has been excellent for welded merchant ships built during recent years of steel material comparable to that of ASTM Specification A131-53T. With adequate improvements in steel material for welded bridge construction in continental Europe, their difficulties seem to have been solved.

Bridge engineers have a choice between two new ASTM specifications for structural steel, for any particular application, A131-53T, specifically designated for ship structures, and A373-54T, Structural Steel for Welding, Tables 2 and 3. They are quite comparable in price. The new AWS bridge specifications will likely mention only A373-54T. It does not assure quite as favorable chemistry for the greater thicknesses of material as the A131 ship steel, nor does it provide definitely for fine grain steel making practice for material over one inch thick, as A131 does. However, A373 is more readily and generally available. The selection of either of these steel materials definitely characterized the solution of the brittle fracture problem as one having for its objective the prevention of *initiation* of brittle fractures.

The better assurance of good notch ductility afforded by either of these specifications, as compared with the older A-7 specification is well worth the small additional cost, in the case of outdoor structures subjected to rather low winter temperatures.

The only substantial additional cost for these newer steel materials is in the rather small range of thicknesses, over 1 in. through 1½ in. This is because the provisions of the newer steel specifications, in effect assure the furnishing of a fully-killed steel for all thicknesses over one inch. For A7 steel it is the common practice of steel mills to furnish a fully-killed steel only for thicknesses greater than 1½ in. Recent research and experience indicate that for some kinds of service conditions such as those for bridges and ships, the change from a semi-killed steel to a fully-killed steel should be made at 1 in. rather than 1½ in. thickness.

Experience as well as research has provided sufficient information to give designers quite a good conception of the types of details that may be used without introducing intolerable constraint or notch effects. Some details that have been discussed in this paper appear to be borderline cases for toleration in a bridge constructed of ordinary carbon structural steel such as ASTM-A7. The use of one of the newer steels A131 or A373 increases the margin of safety.

Likewise experience and prototype tests indicate that it should not be as difficult as it has been supposed by some researchers, to control the welding of one of these newer steels so as to prevent the occurrence of sizable crack-like flaws, which some writers have assumed to be unavoidable.

One result of faulty detailing of parts for a welded structure has perhaps not been stressed as much as it merits. Grossly defective welds or welds of inadequate size and shape for the stress imposed upon them sometimes result from faulty details of design. Sometimes this is because of poor accessibility for welding, and sometimes because of carelessness in failing to detail weld grooves properly. This condition has also resulted at times from welding being introduced by workmen at places where none was intended or needed, but where the weld is in a position to participate in stress.

The hatch corner design of the Liberty Ship has been criticized widely for its geometric shape and arrangement which resulted in compounded notch effects and stress concentrations. It is not so well known that when a small group met to study the design details of the hatches of the Victory Ship, one of the important objectives was to devise an arrangement that would eliminate the lack of accessibility

for good welding which had resulted in seriously defective welding at the hatch-coaming intersections, in way of the deck plating and deck-plate doublers of the Liberty Ships.

A weld that is greatly deficient in either quality or cross-sectional area may crack during early service life, if not during the actual construction operations, and thus provide a point for initiation of fracture. The outboard ends of the transverse butt welds in the deck plating of some ships afforded good examples of this behavior. The groove was prepared for automatic welding, with a very wide root face, but the last few inches were welded manually without first adapting the groove to procedures for manual welding. A grossly deficient weld area resulted.

In one case of a temporary repair, where a short local fracture had occurred in the shell of a tank, holes were drilled at the ends of the fracture, according to instructions, and the area was covered with a reinforcing plate, welded in place. It was not long before a more serious fracture occurred in the same location. An examination showed that an overzealous welder evidently had taken it upon himself to deposit a seal bead along the original crack. To make matters worse, the holes that were drilled as crack-stoppers had been made very small.

Of course, the shell plating beneath the doubler would tend to share the applied load stress. With such a small weld area, as that provided by the seal bead, and with severe stress concentration at its root, the seal bead had cracked. Since the steel was cold, the fracture could easily by-pass the small drilled hole and propagate into the shell plating.

It should be emphasized that the presence of crack-like flaws resulting from fabrication operations is one of the most important factors in the initiation of brittle fracture. They may result from punching, shearing, flanging, friction sawing, and other mechanical operations, or from welding. For welded joints, proper control of welding procedures is relied upon to eliminate such flaws.

A good deal of information has been gained from research and experience on details of design and control of workmanship. The application of such information, together with the use of suitable structural steel material of controlled chemistry would seem adequate to prevent initiation of brittle fracture in welded bridges and similar structures. This is the solution adopted by the AWS committee on specifications for welded railway and highway bridges, in preparing a revised edition of its specifications that is now nearing completion.

As in other kinds of construction, details of design can be covered only by general provisions in the specifications. These must be supplemented with good design judgment based on experience and research.

The workmanship provisions of previous editions of the AWS bridge specifications are being revised to provide more rational and specific requirements for controlled, standardized welding procedures. These include requirements relating to the use of preheat or low-hydrogen electrodes which are patterned quite closely after the admittedly conservative provisions tabulated in "Weldability of Steels."[15]

With the use of one of the newer, quite inexpensive carbon steel materials that have been mentioned, reasonable care in designing and in adhering to construction specifications is relied upon to provide safe welded bridges, merchant ships and other similar engineering structures. Experience thus far, extending over quite a number of years, together with the results of tests of prototypes of structural members, indicates that this solution should be effective and adequate.

REFERENCES

1. Ruzek, J. M., Knudsen, K. E., Johnston, E. R., and Beedle, L. S.. "Welded Portal Frames Tested to Collapse," *The Welding Journal*, 33(9), Research Suppl., 469-s to 480-s (Sept. 1954).
2. Shank, M. E., "Brittle Failure in Carbon Plate Steel Structure Other Than Ships," Welding Research Council Bulletin Series, No. 17, Jan. 1954.
3. Osborn, C. J., Scotchbrook, A. F., Stout, R. D., and Johnston, B. G., "Comparison of Notch Tests and Brittle Criterion," *The Welding Journal*, 28 (1) Research Suppl., 24-s to 34-s (1949).
4. Robertson, T. S., "Propagation of Brittle Fracture in Steel," *The Journal of the Iron and Steel Institute*, 175 (Dec. 1953).
5. Pellini, W. S., "Notch Ductility of Weld Metal," publication pending by Welding Research Council (1955).
6. Pellini, W. S. and Eschbacher, E. W., "Ductility Transition of Weld Metal," *The Welding Journal*, 33 (1) 16-s (January 1954).
7. Grover, L., "Design and Research," and discussion by Lamb, G., "Symposium on Welded Structures," Paper 2729, *Trans. ASCE*, 120, 103 (1955).
8. Armstrong, D. B., "Canadian Welded Girder Bridge Practices," Proceedings of the 30th Annual Meeting of the Highway Research Board, National Research Council (January 1951).
9. Graf, O., "Tests of Welded Bridge Girders," *The Welding Journal*, 20 (3), Research Suppl. 138-s to 148-s (1941).
10. Dunoyer, A., "La Soudure dans la Construction Métallique," Annales de l'Institute Technique du Batiment et des Travaux Publics, Paris, No. 69, (September 1953).
11. Welding and Metal Fabrication, London, England, January 1955.
12. Krefeld, W. J., and Anderson, G. B., "Transition Temperatures of Structural Steel Beams with Butt-Welded Splices," *The Welding Journal*, 32 (11), Research Suppl., 538-s to 576-s (1953).
13. Tör, S. S., Ruzek, J. M., and Stout, R. D., "The Effect of Fabrication Processes on Steels Used in Pressure Vessels," *The Welding Journal*, 30 (9) 446-s, (September 1951).
14. Harris, L. A., "The Effect of Edge Conditions on the Tendency Toward Brittle Fracture of Structural Steels," a report of the Department of Civil Engineering, University of Illinois, July 1954.
15. Stout, R. D. and Doty, W. D., "Weldability of Steel," Welding Research Council (1953).

GEORGE J. RASCHER

THE FIREMAN ABOARD MODERN COMBATANT SHIPS

THE AUTHOR

graduated from the United States Merchant Marine Academy, Kings Point in June, 1951, receiving his Bachelor of Science degree and Third Assistant Engineer's license. Following his graduation, the author sailed actively as a marine engineer aboard ships of the merchant marine until he entered into active naval service in March, 1953 as a Lieutenant Junior Grade. During his tour of active naval duty, the author served as engineering officer of the USS Douglas H. Fox (DD 779) for a period of eighteen months. When he was released to inactive duty in February, 1955, he assumed a position with Hagan Corporation as a field engineer. In this capacity, he has been engaged for the past six months in setting up the automatic boiler combustion and feedwater control systems of the USS Saratoga (CVA 60).

WHEN STEAM became the main motive power for combatant naval ships, there became an immediate necessity for the control of certain boiler processes. In the beginning, fuel and air for combustion and boiler feedwater were controlled manually.

Manual feedwater control, as first used, is similar to single-element feedwater control which eventually evolved. A watertender was the sensing device in a manual control system; the watertender was also the computing device in this system. A manual feedwater check valve was the controlling device.

As boiler drum level varied, the watertender would visually note the magnitude and direction of the change. In his mind, he would then decide in which direction and how much the check valve position should be changed to again bring drum level back to its normal height.

Using the same factors as in manual control systems, the first automatic boiler drum level controller was designed. This system was relatively simple. A float was used as the sensing device for drum level variations; the float was connected mechanically to a feedwater control valve by appropriate linkage. As drum level varied, the float would seek a different height, and through the mechanical linkage the feedwater control valve would be repositioned. Therefore in this system, it is apparent that feed valve position is different for each boiler drum level. For each feed valve position the feeding rate of the boiler would vary. It can be easily seen that in order to change the feeding rate of the boiler, drum level had to vary from normal. Drum level would be normal at only one feeding rate. This resulted in erratic control, and in addition, float control was impractical for practical boiler steaming pressures.

A more modern single-element feedwater control system is still widely used. This system employs a thermal-hydraulic regulator. In the system a two-compartment generator is employed in conjunction with a feedwater regulating valve.

The generator is installed at normal boiler drum operating level. The generator has an inner tube which is filled with a liquid, generally distilled water. The inner tube is surrounded by an outer chamber which is exposed by proper connections to the steam and water spaces of the boiler drum. The inner tube is sealed to atmosphere at one end and attaches to a diaphragm chamber or bellows of the feedwater regulating valve at the other end.

Heat is transferred from the outer to the inner tube of the generator. The amount of heat transferred is dependent upon the water level within the boiler drum. Decreasing boiler drum level brings about a corresponding decrease in the water level in the outer tube of the generator. As the water level in the outer tube of the generator decreases, there is greater thermal heat transfer between the outer and inner tubes. Since the inner tube is sealed to

atmosphere, the liquid in the inner builds up hydraulic pressure. This pressure acts on the bellows or diaphragm of the feedwater regulating valve to open the valve attempting to return boiler drum level to its normal height. Conversely, an increase in boiler drum level causes less thermal heat transfer to the inner tube of the generator which in turn closes the feedwater regulating valve, decreases the feeding rate of the boiler, and attempts to return boiler drum level to its normal value.

It can be noted that any feedwater control attempts to maintain the controlled variable at set point. The controlled variable is the quantity being controlled which, in this case, is boiler drum level. Set point is the value at which it is desired to maintain the controlled variable. In feedwater control systems, the set point is normal drum level.

A thermal-hydraulic feedwater regulator generally contains too much proportional band (the amount of change in drum level that causes the feedwater control valve to go through its full stroke) to be satisfactory under all boiler steaming conditions, such as maneuvering and catapulting. Another inherent disadvantage of any single-element controller is that it depends strictly on sensing drum level to position the feedwater control valve. When boiler load increases and the boiler drum swells, a single-element control system closes the feed control valve. Under conditions of boiler drum shrinkage, the feed control valve is opened. Therefore, it is obvious that a single-element control system positions the feed control valve in the inverse direction under conditions of changing steam load.

To compensate for this inverse positioning of the feed control valve, modern feedwater control systems are of the three-element variety. In a three-element control system sensing devices are used to measure all of the variables which affect the feeding of a boiler:

(1) Drum level within the drum
(2) Steam flow from the boiler
(3) Feedwater flow into the boiler

The transmission fluid in a control system may be oil, water or compressed air. Considering all factors —economy, safety, availability and maintenance— compressed air is the most suitable transmission medium for use aboard ships.

In order to measure steam or feedwater flow there must be a method of obtaining a pressure drop in the steam and feed lines. This is normally accomplished by installing an orifice or flow nozzle in the line. In choosing between the two units, efficiency and initial cost are the primary considerations. An orifice installation is initially cheaper, but a greater pressure recovery is obtained with a flow nozzle.

The size of the flow nozzle or orifice is calculated mathematically. By calculation the maximum differential across the orifice or nozzle at maximum flow through the line may be determined. Knowing the maximum differential an instrument can be fitted into the control system to give an air loading signal that is proportional to flow through the line.

The flow element used may be either:

(1) A Differential Transmitter
(2) A Flow Transmitter

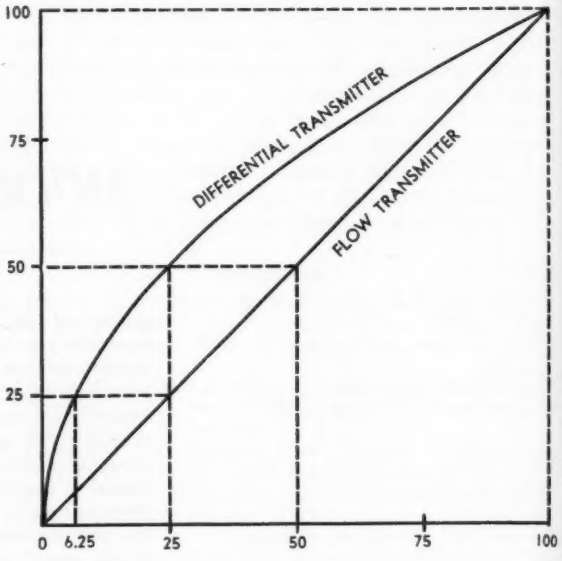


Figure 1

Flow varies with the square root of the pressure drop within the line or across the orifice or nozzle. A differential transmitter has an air output that varies directly with the differential; this produces a non-linear curve as indicated in Figure 1. One instrument manufacturer incorporates feedback and a square root cam in a differential transmitter to convert it to a flow transmitter. This or any other flow transmitter has a pneumatic output signal that is linear with flow as indicated in Figure 1.

Since the magnitude of the pneumatic output of a differential transmitter is small at low flows as shown in Figure 1, the steam and feedwater flow influence in a three-element control system, using differential elements, is correspondingly small at low flows. In actual practice, this type of system gets so little influence from steam and feedwater flow at low boiler steaming rates that it acts as a single-element control system at low rates. For this reason flow elements are preferred in control systems. In addition, since a flow transmitter gives a linear output signal with flow, gain and reset times of the computing devices may remain essentially the same throughout the entire operating range of the boiler without decreasing system sensitivity at any boiler load. When differential elements are used in the

control system, sensitivity must be sacrificed at low boiler steaming rates to maintain system stability at high steaming rates.

However, differential transmitters are always used as drum level transmitters because for this application a pnuematic output which is linear with differential is desired. A drum level transmitter is calibrated to go through its full output range while the drum level varies through its normal working range or that portion of the drum which is visible in the drum gauge glasses. In order to function, the drum level transmitter is connected to the boiler drum through appropriate lines as indicated in Figure 2. One line is connected to the water space of the drum, and the other is connected to the steam space. The drumlevel transmitter would be calibrated for range "x". The line which comes from the steam space is the reference leg for the instrument because steam condenses to water in this line. Once the steam is condensed in this line, the pressure exerted by this line on the transmitter is always constant. The difference of height sensed by the transmitter is the difference between the reference leg and actual drum level. For this reason, a drum level transmitter senses a greater differential with decreasing level within the drum. Thus the transmitter sends an increasing pneumatic signal with reducing drum level.

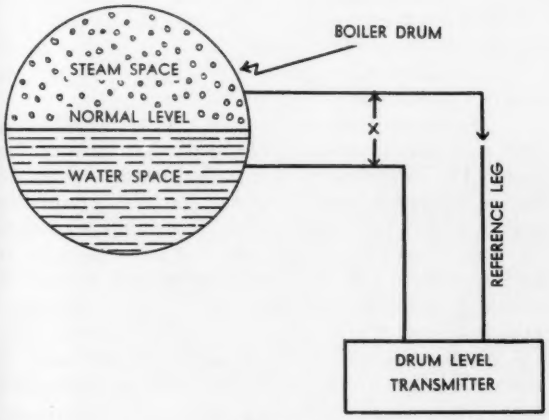


Figure 2

The computing devices employed in control systems may differ in construction. But to give effective control response, the construction must allow for the following:

(1) Mathematical computation of pneumatic input signals to give a pneumatic output signal equivalent to the mathematical answer.

(2) Adjustability of or proportional band to make the system as sensitive as possible without the introduction of instability as is reflected by hunting in the system.

(3) A method of providing reset action in the system in order to maintain drum level at its normal steaming level, regardless of the boiler steaming rate.

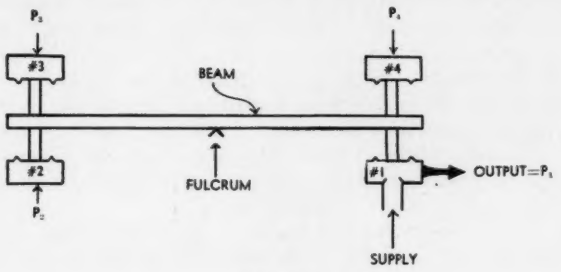


Figure 3

A totalizer is a typical computing device. A brief study of a standard totalizer reveals that its features of construction are ideal to give the control response which is described in the preceding paragraph. Figure 3 shows schematically the construction and operation of a standard totalizer.

The totalizer, depicted in Figure 3, consists of four, flexible, non-metallic diaphragms arranged in opposing pairs of two. The two diaphragms of each pair are attached to a common post. The posts act on opposing ends of a beam. The beam is fitted with an adjustable fulcrum about which it is free to rotate.

Input signal pressures are piped to any or all the input loading diaphragms (chambers #2, 3, and 4 of Figure 3). All diaphragms of the totalizer are encased in suitable chambers. Chamber #1 contains an air poppet valve which is supplied with compressed air at the desired pressure. Another connection is made to chamber #1 to allow the output of the totalizer to go to other elements of the control system. The poppet valve has two seats which operate between inlet and exhaust ports of the valve assembly.

The operation of the totalizer depends on the force balance principle. The force acting on each diaphragm is equal to the pressure acting on each diaphragm times the area of the diaphragm. Since the area of each diaphragm is one square inch, the force acting on each diaphragm is equivalent to the pressure imposed on it. Since the sum of the clockwise moments equals the sum of the counter-clockwise moments when the beam is in equilibrium, the loading within chamber #1 is the solution of a problem of moments resulting from imposing loading pressures in the other three chambers. If the length of the lever arm to the left of the fulcrum is designated L_1 and the length of the lever arm to the right of the fulcrum is designated L_2, the problem in moments is shown by the following equation:

$$P_2L_1+P_4L_2=P_3L_1+P_1L_2$$

The above illustrates that the totalizer acts as a computing machine.

The adjustable feature of the beam permits the gain of the totalizer to be changed. By moving the

fulcrum to the right of the center, a one psi change in signal pressure in chambers #2 or 3 will cause a change of greater than one psi in the output signal from chamber #1; the magnitude of the signal from chamber #1 under these conditions depends on how far to the right of the center of the beam the fulcrum is moved. In this manner, the gain of a totalizer may be increased to a maximum value that is limited only by the length of the beam.

Conversely, moving the fulcrum of a totalizer to the left of the center of the beam, decreases its gain. In a control system, the highest gain possible is used to make the system as sensitive as possible. The only limit placed on sensitivity is the point at which a system becomes unstable. In an over sensitive system, the final controller will hunt, causing corresponding fluctuations in the set point of the controlled variable. In the case of a feedwater control system, the feed control valve would hunt causing fluctuations in drum level.

Finally, a totalizer may be equipped with reset action. Reset control response is defined as output proportional to the average error times the elapsed time during which the error occurred, or output proportional to the time integral function of the error. Error, as used here, is the difference between the set point and actual value of the controlled variable.

Figure 4 illustrates a totalizer fitted with reset action. The output signal is led through a needle valve which is throttled to determine the rate at which reset action occurs. When the input signals, P_2 and P_3, are equal, the totalizer is in equilibrium. In an equilibrium condition, the output pressure, P_1, must equal the pressure, P_4. When P_2 and P_3 are unequal, the output signal, P_1, will continue to change until P_2 and P_3 are again equal. This action can be best illustrated by assuming that the value of P_2 is greater than P_3. If this is the case, P_1 will increase; the amount of increase will depend upon the gain of the instrument. However, as the signal P_1 bleeds through the needle valve and volume tank, a further increase in P_1 will result. The rate at which P_1 increases depends upon the speed at which the output bleeds into chamber #4. The output signal will continue to increase until the totalizer is again in equilibrium. This results only when P_2 and P_3 are equal. This is a practical example of reset response as employed in control systems.

Before considering a typical three-element feedwater control system, the controlling device should be considered. The controlling device is normally a balanced valve that is either, bellows or diaphragm-actuated, and spring loaded. The valve is normally fitted with feedback to insure positive positioning proportional to the pneumatic loading signal to the valve. The valve should be fitted with a mechanical means, such as a handwheel, to positively jack the valve open in case of failure to any component of the pneumatic control system which would cause faulty feedwater regulation. The ideal feedwater

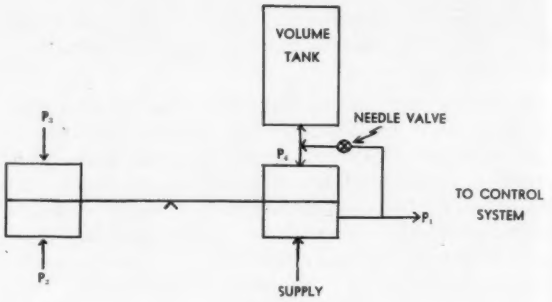


Figure 4

regulating valve is fitted with an auxiliary diaphragm that is used in conjunction with an emergency back-up unit in the event of air supply pressure failure.

The back-up unit upon air failure, causes hydraulic pressure to be imposed on the auxiliary diaphragm of the feedwater regulating valve. The hydraulic pressure is generated by the generator previously described. Thus, in the event of air failure which makes the three-element pneumatic control system inoperative, boiler feedwater continues to be controlled by a single-element, thermal-hydraulic control system.

Now that the individual sensing, computing and controlling components of a three-element control system have been discussed, the operation of the entire system should be considered. Figure 5 is a schematic drawing of a typical three-element system as installed aboard ship or in a stationary plant.

As shown in Figure 5, an orifice is installed in both, the main steam and the feedwater lines of the boiler. The orifice must be installed in the feedwater line at a point where all of the feedwater into the boiler passes. The same is true in the case of steam line orifice installation, except that blowdown from the boiler cannot be practically measured as part of normal flow from the boiler. In cases where a continuous blowdown is employed in boiler operation, it is usually desirable to have the three-element system compensated for blowdown. Since combatant ships do not generally employ continuous blowdown, Figure 5 shows an uncompensated system.

In the illustrated system it is assumed that full boiler capacity is 200,000 lb/Hr. Steam and that feedwater flow orifice calculations would provide for a known differential pressure equal to full boiler capacity. This is an essential feature of the system. If meters were calibrated to give maximum output equivalent to full boiler load, the system would give unsatisfactory control at maximum boiler load since the system would function as a single-element control system, receiving only corrective pneumatic loading signals from the drum level transmitter.

A wide range of pneumatic signal ranges are employed by instrument manufacturers. A 0-30 psi sys-

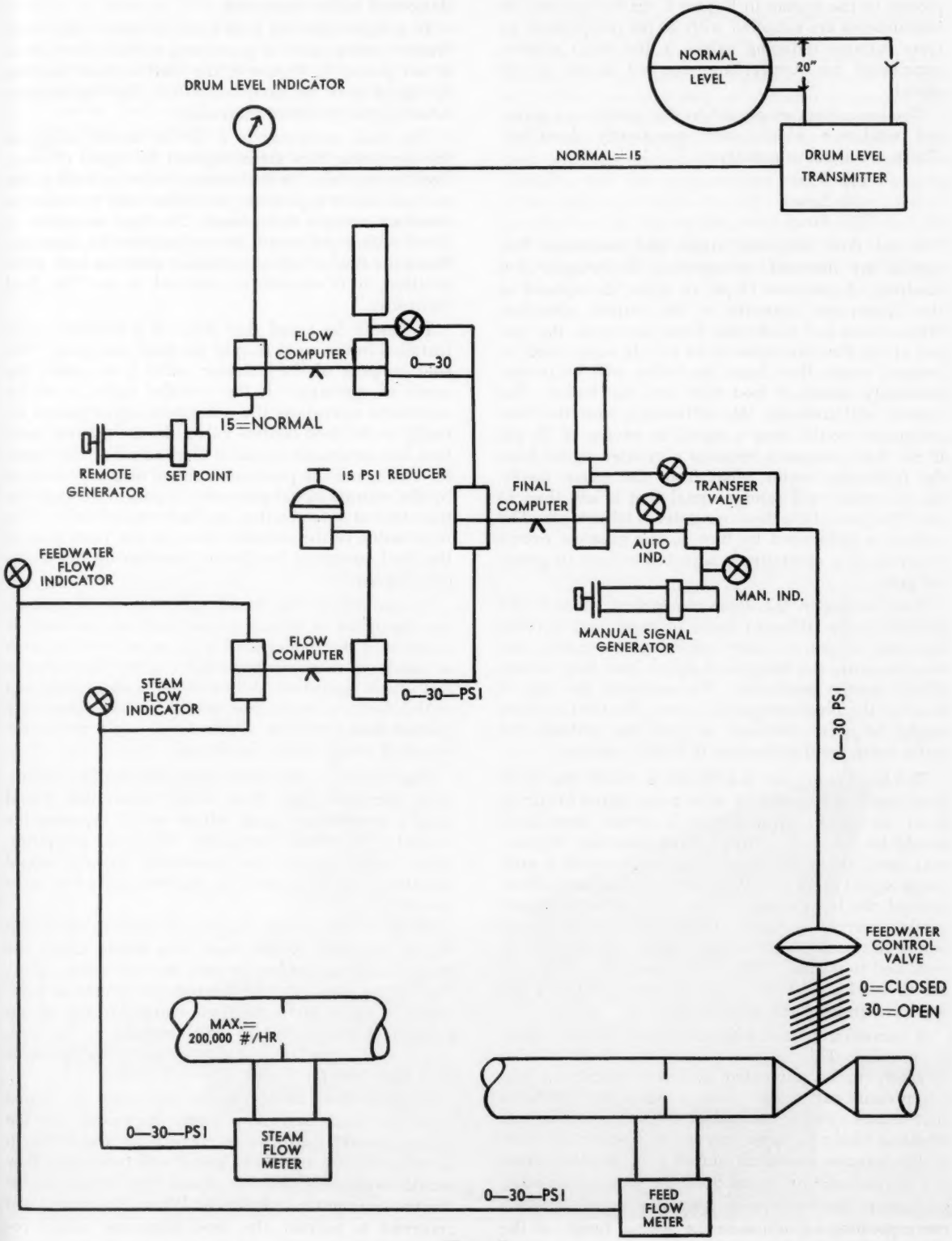

NORMAL
LEVEL
20"
DRUM LEVEL INDICATOR
NORMAL=15
DRUM LEVEL
TRANSMITTER
FLOW
COMPUTER
0—30
15=NORMAL
REMOTE
GENERATOR
SET POINT
15 PSI REDUCER
FINAL
COMPUTER
TRANSFER
VALVE
AUTO
IND.
MAN. IND.
MANUAL SIGNAL
GENERATOR
FEEDWATER
FLOW
INDICATOR
FLOW
COMPUTER
STEAM
FLOW
INDICATOR
0—30—PSI
0—30 PSI
FEEDWATER
CONTROL
VALVE
0=CLOSED
30=OPEN
MAX.=
200,000 #/HR
0—30—PSI
STEAM
FLOW
METER
0—30—PSI
FEED
FLOW
METER

Figure 5

tem is one of the more commonly used; it is employed in the system in Figure 5. In the system, the instruments are supplied with 30 psi compressed air from suitable reducing valves in the ship's system; compresed air supply lines are not shown in the sketch.

The computers employed in the system are standard totalizers which were previously described. There are three computers:

(1) Flow
(2) Level
(3) Final

In the flow computer, steam and feedwater flow signals are imposed on opposing diaphragms of a totalizer. A constant 15 psi air signal is imposed on the diaphragm opposite to the output chamber. When steam and feedwater flows are equal, the output of the flow computer is 15 psi. If boiler load increases, steam flow from the boiler will be instantaneously ahead of feed flow into the boiler. The meters will indicate this difference, and the flow computer would send a signal in excess of 15 psi. If the flow computer receives a greater signal from the feedwater meter, than from the steam meter, the computer will send a signal that is less than 15 psi. The gain of the flow computer is adjustable. The system is influenced by flow to the greatest extent when the flow computer is adjusted to have its greatest gain.

Two boilers in the same machinery space might possibly have different rates of reset and varying amounts of gain in their individual computing devices because the lengths of signal lines in a system affects system sensitivity. For example, the rate of reset of the level computer in two identical systems might be quite different to give the systems the same drum level influences in both instances.

The level computer is a device in which the drum level signal is opposed by a set point signal for drum level. In the system in Figure 5, normal drum level would be ten inches from either extreme. At normal level, the drum level transmitter sends a mid-range signal of 15 psi. When drum level varies from normal, the level computer becomes unbalanced and sends a correcting signal. The level computer would continue to send a correcting signal until drum level returned to normal. This would cause the transmitter to send a mid-range signal again, bringing the level computer back into balance.

A manual means of adjusting drum level set point is provided. This feature is desirable where a boiler is likely to be subjected to swell conditions that might cause carryover. Catapulting is one operation that causes swelling in modern high-pressure boilers which is likely to cause carryover. Therefore, when a ship intends to launch planes it is accepted practice to increase the signal from the manual set point generator. An increase of set point signal causes a corresponding reduction in the normal height of the water within the drum. Generally, if normal drum level is reduced by a small amount, there will be no danger of boiler carryover.

In a three-element feed control system, the feedwater control valve is positioned to hold drum level at set point. In Figure 5, the feed control receives its signal from the final computer. The valve opens when signal pressure increases.

The final computer is a device which compares the correcting flow signal against the signal of drum level to position the feed control valve to hold at the normal value regardless of boiler load in order to maintain normal drum level. The final computer is fitted with reset action to accomplish this function. Since the final computer actually controls feed valve position, it is sometimes referred to as "the final controller."

It should be noted that there is a transfer valve installed in the reset loop of the final computer. The sole purpose of the transfer valve is to select the mode of operation. If the transfer valve is set for automatic operation, the automatic signal passes directly to the feed control valve. In the manual position, the automatic signal is blocked-off at the transfer valve, and the pneumatic signal that is generated by the manual signal generator is passed through the transfer valve to position the feed control valve. The installation of the transfer valve in the reset loop of the final computer facilitates transferring the method of operation.

In considering the overall operation of the system, one condition of changing load will be assumed. It is assumed that boiler load is increased rapidly, such as results in maneuvering main engines from stop to full astern operation. When boiler load is increased in this fashion, steam flow would be instantaneously greater than feed flow. Drum level would rise as the result of swell within the drum.

The output of the flow computer would immediately increase. The drum level transmitter would send a decreasing signal which would increase the output of the level computer. The final computer, upon receiving the two increasing signals, would maintain the feed control in approximately the same position.

Next, the drum level would decrease as the result of the increased boiler load; this would cause the drum level transmitter to send an increasing signal. This signal would reflect through the system as a decreasing signal from the level computer and an increasing signal from the final computer. In turn, this would open the feed control valve and increase feed flow into the boiler.

As feed flow into the boiler increases, the signal from the feedwater flow meter increases. As the system would continue to correct for the error in drum level, the signals of steam and feedwater flow would eventually become equal; this would return the flow computer to balance. When the drum level returned to normal, the level computer would return to balance. When balance is reached in these

two computers, the final computer will also be returned to balance. The output from the final computer will remain the same, maintaining the same feed control valve position, until the system is again disturbed by transient boiler conditions. For each changing boiler condition, the three-element system will correct feed control valve position to bring the system back into balance.

Now, there are obvious advantages gained by using a three-element control system. The advantages are clearly shown in the previous discussion:

(1) Greater speed of response—This is gained by initiating feedwater flow changes from steam flow changes.

(2) Greater stability—This is gained by balancing feedwater flow against steam flow in an instrument that provides a means of adjusting gain.

(3) Accuracy—This is achieved by employing a reset computing device for the drum level influence.

These advantages of three-element control are operating prerequisites in modern high capacity boilers which are operated at elevated pressures and temperatures with little water storage capacity in the drum. The design of modern naval boilers coupled with aggravated conditions of boiler load fluctuations and the requirement for more exacting drum level control has brought about more rigid requirements for the modern watertender. This is the primary reason that the watertender aboard modern ships is generally a three-element control system.

Courtesy New York Shipbuilding Corp.

U.S.S. CANBERRA (CAG 2) The *Boston's* sister ship as she left the builder's yard for acceptance trials.

DESIGN CONSIDERATIONS FOR NAVAL GAS TURBINES

ACKNOWLEDGEMENT

G. L. Graves, Jr., of the Gas Turbine Branch of the Bureau of Ships, is the author of this paper which was presented at the American Society of Mechanical Engineers' Gas Turbine Power Conference in Washington, D. C., 16-17 April, 1956.

INTRODUCTION

IT HAS BEEN previously reported that the engine characteristics which have been most successfully demonstrated by the naval gas-turbine engine are its light weight, small size and reliability of starting (1, 2). Quick positive starts have been possible even when operating the engines in the Arctic. The most important characteristic of the naval gas turbine is its ability to pack more power in less space than any other prime mover. The utilization of naval gas turbines for booster and emergency applications is becoming a reality.

More conservative design techniques have resulted in industrial and marine gas turbines with excellent maintenance characteristics (3, 4). In fact Shell Oil Company's experience with the *Auris* tanker has looked so promising that they are now planning on speeding up their entire gas-turbine ship-installation program. Gas turbines will provide the sole means of direct mechanical drive for the *Auris* propellers. The U.S. Maritime Administration has also embarked on a program for the conversion of several Liberty ships (5). These very vigorous programs will do a great deal to establish the gas turbine as a marine prime mover. In these installations the power plant is designed for continuous full-power operation, good maintenance and a minimum of down time for overhaul. Engine specific weight is not particularly important.

The purpose of this paper is to reflect on the design considerations for a naval gas turbine in terms of duty-profile operating requirements and the basic parameters of engine design. The combatant naval gas turbine might be defined as a compromise between aircraft and industrial gas-turbine-design concepts. The degree of compromise should reflect the importance of weight, performance and engine life for the specific application.

APPLICATIONS

Emergency Applications

Modified aircraft design concepts applied to small gas turbine engines have resulted in engines weighing about one pound per horsepower. Engines can advantageously be used in applications where any of the following are important:

1. Light weight and low volume where it is desired to operate without the hazards of gasoline.
2. Rapid positive starts even under conditions of subzero temperatures.
3. Availability of hot gas for deicing or generation of oil fog.
4. Availability of compressor bleed air for pneumatic power.

A lightweight gas turbine can be used simultaneously for several applications. An emergency gas turbine-driven fire pump has successfully operated as a deicer in the Arctic as well as a smoke generator.

For this type application aircraft-design concepts have proven to be very satisfactory. Engines have had good reliability within the short operating life for which they were designed.

Special and Standby Applications

The transition from an emergency prime mover to a standby or continuous-duty application will depend on programs aimed at improving the performance of engine components. The first big step is to establish the reliability of engine components and accessories by operating the engines in the Fleet under actual conditions. In one instance approximately 200 engines have been installed in small boats and are now in operation by the fleet. This special application represents an invaluable opportunity to evaluate the special fleet problems associated with this new form of prime mover. Obtaining statistical data on a large number of small engines is a great deal less expensive and statistically more valid than obtaining similar data on one or two large engines.

The next step is to establish research projects to

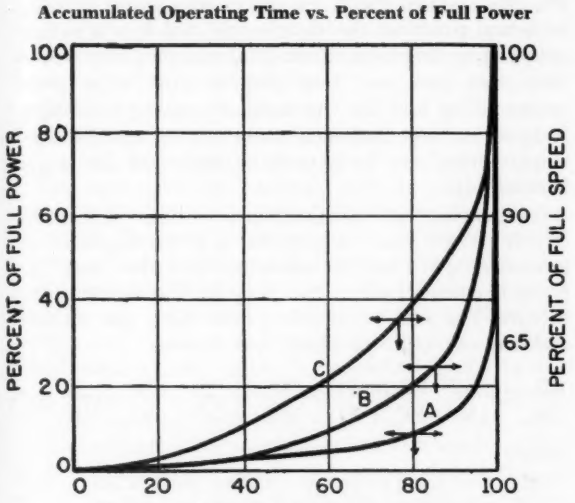


Figure 1. Accumulated Time in Percent.

improve the performance of engine components. These programs should emphasize improved engine life and reduced fuel consumption. As the fuel consumption and hours between engine overhaul are improved the applications for the engines will increase to include standby applications. Reductions in engine costs are dependent on broadening the application spectrum for a given engine.

For certain special applications such as the propulsion of high-speed boats the lightweight characteristics of gas turbines are ideal. The type or combination of prime movers which will give the high speeds required for a continuous period of time will justify the power plant design evolved. Where the application has a part-load and a full-load cruising condition, a multiengine power plant is desirable. The power plant combination will depend on the ratio of time spent at the two cruise conditions and on the power ratios. Where the ratio of full power to part-load power is no greater than three or four to one, a combination of simple-cycle gas turbines would be practical. For applications where the power ratio is greater than four to one, a diesel engine might be used for the base-load power in combination with gas-turbine booster engines.

Booster or Peak-Load Engine Applications

Gas turbines are being evaluated and will be used in combination with base-load prime movers as booster or peak-load engines. Curve A of Fig. 1 illustrates the duty profile of a type of naval vessel (7). A large percentage of installed power is required only 1 per cent of the time or less. A considerable weight saving appears to be possible if lightweight gas turbines are used to supply the power increment which is required less than 1 per cent of the time. The real weight savings are dependent on the weight difference between the all-steam plant and the base-load steam plant with the boosters. Fig. 2 illustrates the specific weights of various power plants versus horsepower per shaft. The all-steam plant would have a lower specific weight than the base-load steam plant for a combined gas-and-steam-turbine installation. For certain power ranges, this differential might negate the weight advantage of the booster gas turbine. It would be preferable to consider a base-load gas turbine for the lower power base-load applications where the specific weight of the steam plant is rather high.

The booster gas-turbine concept is dependent on the lightweight characteristics of aircraft design. Without the rapid starting characteristics of aircraft designs, the booster concept would not be possible. By designing for an engine life of 500 hr, the booster engine will have a corresponding ship life of about 50,000 hr.

There is a great deal to be said for using high production-rate jet engines as replaceable short-life gas-producing sections and developing longer life power turbines and gears for the output section. This approach is satisfactory if the basic aircraft engine is reliable within its operating life at the derated naval rating. Engine development cost would also be a great deal less by using aircraft gas producers.

The Navy special fuel used in boilers could probably be burned in 500-hr life booster engines of this type provided the following things were done:

1. Develop a residuals-fuels combustor interchangeable with aircraft-engine combustor.
2. Develop fuel pumps and accessories for heavy fuels which would be interchangeable with aircraft designs.
3. Reduce turbine-inlet temperatures to about 1300° F.

A turning device to rotate gas producer-engine shafts to prevent ball and roller bearings from brinelling would be required for periods when engines were not in use. The power-turbine output section would include sleeve bearings.

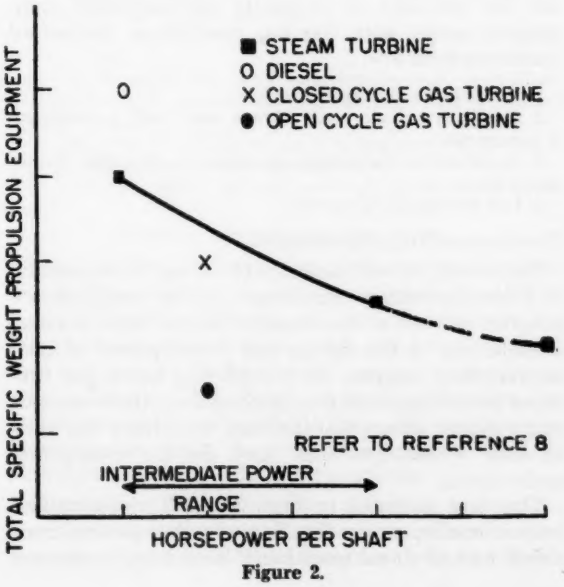


Figure 2.

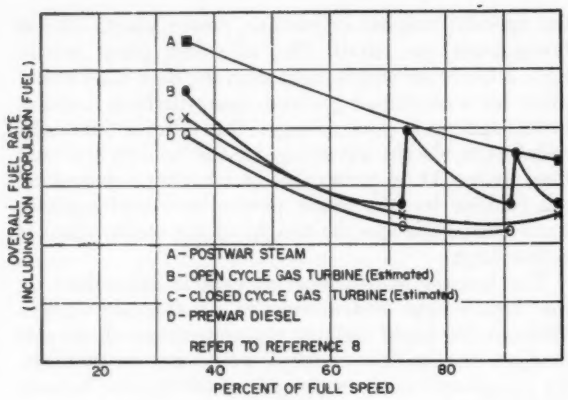


Figure 3.

To keep booster-engine air rates to a minimum a low pressure-ratio engine is preferred. A single-spool engine with a free-power turbine is preferred for simplicity.

The free-power turbine would be matched to the peak-load propeller characteristics. The range of peak-load RPM would depend on the division of power between base-load and booster engines. The booster engines would probably clutch into the base-load-engine propeller-drive shaft.

The booster turbines are not expected to be operated more than a small fraction of the total operating time; however, the quantity of fuel required to be bunkered for its use will depend on the total number of booster operating hours for a given mission.

For applications, where required booster bunkering capacity is a minimum and it is feasible to use diesel fuel, the booster engine eventually could be operated at temperatures of 1600° F and pressure ratios of seven or eight to one for minimum air rate. The pressure ratio might be compromised at six to one for the sake of simplicity and improved compressor surge and starting conditions. Important considerations are:

1. Reliability within operating life.
2. Simplicity—single push button start with a minimum of accessories.
3. Rapid starts—surge-free operation of compressor during acceleration.
4. Low air rate at full power.

Continuous-Duty Applications

Eventually, it will be desirable to apply the results of Fleet operating experience on the small-engine program as well as the results of larger experimental installations to the design and development of continuous-duty engines. In considering naval gas turbines for continuous-duty applications, there are not many places where gas turbines now have the very obvious advantages that they do for emergency applications.

The data included in Figs. 2 and 3 was obtained from an earlier paper (8). Some further general considerations of naval machinery have been presented (9). The data in Fig. 3 are shown as a series of lines. In actual practice, the data would fall into a rather broad operating region and fuel consumption would vary with time and from ship to ship for a given power plant and for the same operating condition. Indications are that this variation in specific fuel consumption can be kept to a minimum for a gas turbine (4).

Cruising-radius requirements often determine which power plant is used for a given application; however, space and maintenance are also very important considerations and may be the determining factors. The ratio of cruising radii for a gas turbine and any other power plant is as follows:

$$\frac{CR_{gt}}{CR_p} = \frac{(NF+PF_p)}{(NF+PF_{gt})} \frac{FD_{gt}}{FD_p} \frac{HV_{gt}}{HV_p} \left[FT + \frac{K}{FD_{gt}} (W_p - W_{gt}) \right]$$

where

CR_{gt} = cruising radius gas-turbine installation
CR_p = cruising radius some other power plant
NF = nonpropulsion fuel (1b/hr)
PF_{gt} = gas-turbine propulsion fuel for X knots, (1b/hr)
PF_p = other power plant propulsion fuel for X knots, (1b/hr)
FD_{gt} = gas-turbine fuel density
FD_p = other power plant fuel density
HV_{gt} = heating value of gas-turbine fuel
HV_p = heating value of fuel for other power plant
FT = fuel tankage, cu ft
W_p = weight of other power plant (total machinery)
W_{gt} = weight of gas turbine (total machinery)
K = percentage of gas-turbine weight saving which would be converted to fuel tankage

If feedwater is eliminated, feedwater-tankage space can be used for fuel storage.

There appears to be a very definite application for a combatant-type naval gas turbine in the intermediate power range. The following conclusions can be drawn from Figs. 2 and 3 relative to vessels in the intermediate power range:

(a) Nonpropulsion fuel has an appreciable effect on the total fuel consumption at part-load cruising conditions. Where the ration of nonpropulsion fuel to propulsion fuel is significant the slope of the total fuel-consumption curve is quite a bit different from that of the basic prime mover.

(b) The total fuel consumption of an open-cycle gas turbine could be better than a steam turbine and almost as good as a diesel. The percentage effect of the ratio $(NF + PF_p)/(NF + PF_{gt})$ closes the gap between the fuel-consumption curves.

(c) A good way to increase cruising radius is to increase fuel tankage. This is especially true for part-load conditions where nonpropulsion fuel is a large percentage of total fuel consumption.

(d) The total specific weight for an open-cycle gas-turbine propulsion-machinery plant is estimated to be approximately one half the total machinery weight of the other plants considered.

(d) The weight savings associated with using lightweight gas turbines could be used to carry extra fuel. The percentage of weight saving which could be converted to fuel tankage would be dependent on ship-installation studies for a specific vessel.

About one-fourth of the estimated total open-cycle gas-turbine machinery weight is the basic gas-turbine engine. By doubling the specific weight of the basic engine, the total machinery weight of the open-cycle gas-turbine installation would still be less than five-eighths the total specific weight of other propulsion-machinery installations.

The open-cycle gas turbine may prove to be the most advantageous propulsion-machinery power plant for the intermediate power range. If such a power plant development were undertaken, the naval architect would be required to develop ship designs or conversions which could utilize the weight and space saving resulting from a gas-turbine installation. It may prove to be advantageous to use the weight saving for other purposes besides increasing cruising radii.

In considering the potentials of gas turbines for the intermediate power range, it is concluded that it is not necessary to design the engine for the last ounce of efficiency attainable. The effect of nonpropulsion fuel deemphasizes the importance of efficiency. Neither is it desirable to overemphasize light weight since the basic engine weight is only a small percentage of the total machinery weight. The ability of the gas turbine to pack more power into less weight and space is its main advantage for propulsion in the intermediate power range.

DUTY-PROFILE CONSIDERATIONS

One of the biggest advantages of the gas turbine is its flexibility of design and arrangement. This flexibility enables the design of a power plant which can reflect the relative importance of the duty profile of the application. For combatant vessels, this flexibility results in a possible series of combinations of complex and simple-cycle gas turbines. It is emphasized that a weighting system based on duty profile results in an optimum power plant; however, optimization is dependent on certain limits. The resulting power plant cannot be deficient in any one requirement.

Curves A, B, and C in Fig. 1 represent estimated duty profiles for several classes of vessels.

(a) For all curves shown, a very large percentage of time is spent at part loads. This necessitates designing for good part-load performance.

(b) As a duty profile for a vessel approaches curve A, part-load performance becomes even more important. It is for this type of duty profile that booster engines show up most advantageously. Where a base-load plant below 15,000 shp is required, gas turbines might be used in order to take full advantage of the booster-engine weight savings.

(c) For curves B and C and a power plant power requirement from 5000-15,000 shp, the power plant might consist of a number of simple-cycle gas-turbine engines coupled together with a heat exchanger mounted on one of the engines. The gas-turbine power plant for curve B might consist of three engines; the power plant for curve C might consist of two engines.

The multiengine concept has been discussed previously and there is nothing new or revolutionary about the idea (10). In fact, the idea has been considered for marine plants (11).

The size effects of gas-turbine engines are important. It has been noted (2) that the specific weight of gas-turbine engines increases with power in accordance with the following formula:

$$\frac{\text{lb/HP of 15,000-HP engine}}{\text{lb/HP of 5000-HP engine}} = \sqrt{\frac{15{,}000}{5000}} = 1.73$$

Whereas this formula is only a rough approximation, it does indicate a definite trend. A multiengine plant consisting of three 5000-hp engines would weigh less than a 15,000-hp engine. The following advantages would result from utilizing a multiengine plant consisting of a number of basic engines in lieu of a single engine:

(a) Improved reliability at the cruising condition. Even if one engine is down, top speed is compromised only about 8 per cent (based on 3 engines for curve B Fig. 1). All engines would be suited for extended period of continuous duty operation.

(b) Ease of overhaul. The smaller the parts, the easier and quicker it is to disassemble and overhaul the engine. Underway maintenance of gas turbines may be feasible for multiple-engine installations.

(c) Smaller size of spare parts. Complete rotor assemblies can be carried as spares for easy maintenance.

(d) Less critical materials.

(e) Weight saving on regenerator. The regenerator can be designed for operation with single-engine mass flow of a three-engine plant. A single-engine plant would have to include a regenerator bypass for full-load operation in order to keep regenerator weight to a minimum.

(f) Smaller rotor forging of multibarrel engine would be more reliable (better quality control in production) and easier to procure than large forgings. (More facilities are available for producing small forgings.)

(g) Multiple-engine plants are more easily divisible into optimum sizes for handling.

(h) Shorter engine length. Length is usually the critical dimension in installation studies.

(i) Better part-load specific fuel consumption especially in the cruising range.

(j) Wider application. The basic engine could be utilized in multiples with and without heat exchangers for a number of applications.

(k) Reduced development cost. It is less costly to develop a 5000-hp engine than a 15,000-hp engine.

(l) Decreased down time of power plant. (Easier to overhaul.)

GENERAL DESIGN CONSIDERATIONS

The maintenance advantages of a combatant naval gas turbine are rather difficult to evaluate at this time. Although reliability and maintenance are related, it is possible to design for good maintenance with long periods between overhauls and yet not have good reliability. Reliability might be defined as inversely proportional to the number of forced outages. Maintenance might be defined as proportional to forced-outage costs plus overhaul costs.

Good reliability is dependent on having a minimum number of engine accessories and components in series; therefore simplicity of design is desirable. Items

like intercoolers and other accessories have to be weighed in terms of complexity added to the engine versus performance gains.

Reliability within the design overhaul life of the engine is the important consideration.

Maintenance is not directly proportional to the overhaul life of an engine. The down time is also important. Engines should be designed for rapid disassembly and replacement of parts. Excessive down time is reflected in increased manpower requirements for engine overhaul and lack of availability of the entire ship investment. It is contemplated that gas turbines will decrease down time for overhauls and thus decrease manpower requirements at overseas bases. They will also increase availability of the military equipment and vessels in which they are installed.

Certain classes of vessels are overhauled about every 15,000 hours. For a vessel with an activity rate of 28 per cent, the basic engine will accumulate about 4000–5000 hours of duty-profile operating time between overhauls. Of this time, perhaps a maximum of 200 hours at full power would be required.

A gas turbine designed for 5000 hours of duty profile operating time or a maximum of 1000 hours at full power would meet present requirements.

For this type of design life (hours between overhauls) modified aircraft-design techniques will certainly prove to be satisfactory.

Engine controls should be kept to a minimum. In the event of an emergency the minimum number of operations should be required to throw the gas turbines on the line. Complicated valving is certainly not desirable.

THERMODYNAMIC DESIGN CONSIDERATIONS

The importance of reliability, maintenance, ease of fabrication and ease of overhauls results in a great many aerodynamic and thermodynamic compromises.

Any thermodynamic study of the general type represented herein only begins to approximate actual engine conditions. The only purpose of such a series of graphs is to indicate generally the relative magnitude of importance of temperature, pressure ratio,

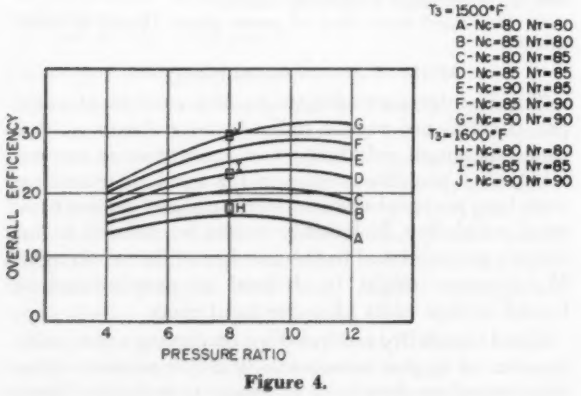


Figure 4.

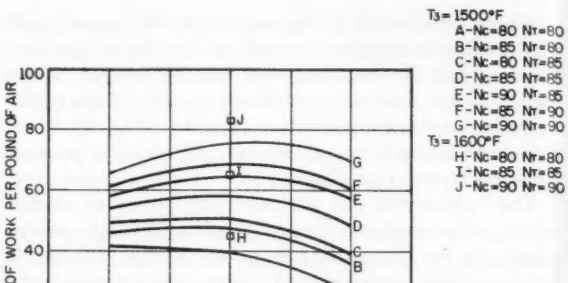


Figure 5.

and component efficiencies on over-all efficiency and air rate. The extent of leakage and duct losses in an engine are appreciable and are not always known.

Figs. 4 and 5 are plots of net work per pound of air and efficiency against pressure ratio (refer to Tables 1 and 2). Depending on the relative importance of air rate and efficiency, the optimum pressure ratio for a given application can be determined. Points H, I, and J are plotted for 1600° F turbine-inlet temperature. It is noted that for a given pressure ratio increasing the temperature increases the work output per pound of air whereas the improvement in

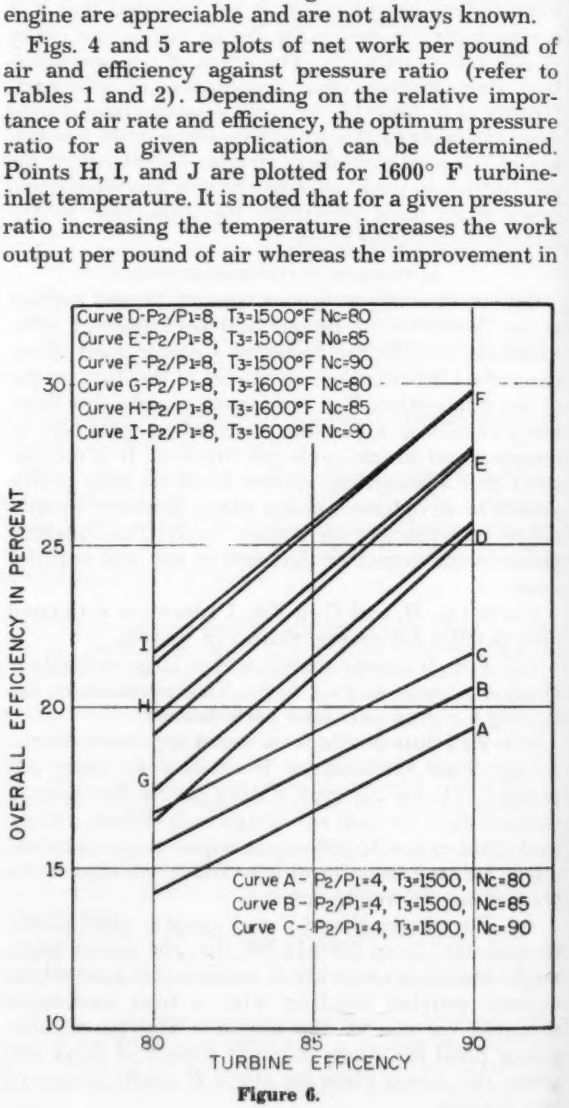

Curve D-P2/P1=8, T3=1500°F Nc=80
Curve E-P2/P1=8, T3=1500°F No=85
Curve F-P2/P1=8, T3=1500°F Nc=90
Curve G-P2/P1=8, T3=1600°F Nc=80
Curve H-P2/P1=8, T3=1600°F Nc=85
Curve I-P2/P1=8, T3=1600°F Nc=90
Curve A - P2/P1=4, T3=1500, Nc=80
Curve B - P2/P1=4, T3=1500, Nc=85
Curve C - P2/P1=4, T3=1500, Nc=90
OVERALL EFFICIENCY IN PERCENT
TURBINE EFFICENCY

Figure 6.

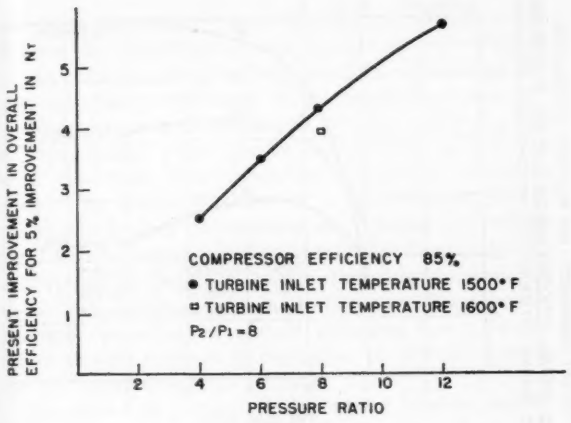

PRESENT IMPROVEMENT IN OVERALL EFFICIENCY FOR 5% IMPROVEMENT IN NT
COMPRESSOR EFFICIENCY 85%
● TURBINE INLET TEMPERATURE 1500° F
□ TURBINE INLET TEMPERATURE 1600° F
P2/P1 = 8
PRESSURE RATIO

Figure 7.

efficiency is small. To take advantage of increased turbine-inlet temperatures it is desirable to go to higher pressures or include a regenerator in the cycle.

Fig. 4 also illustrates the fact that for component efficiencies of the order of 80 per cent, it is not desirable to go to higher pressures.

Fig. 6 illustrates the effect of turbine efficiency on over-all performance. At pressure rations of 8:1, turbine efficiency is a great deal more important than it is at 4:1. For the higher values of component efficiencies, an increase of 100 deg. in turbine-inlet temperature has a very small effect on performance at full power; however part-load power is considerably improved.

Fig. 7 illustrates the per cent improvement in over-all efficiency for a 5 per cent improvement in turbine

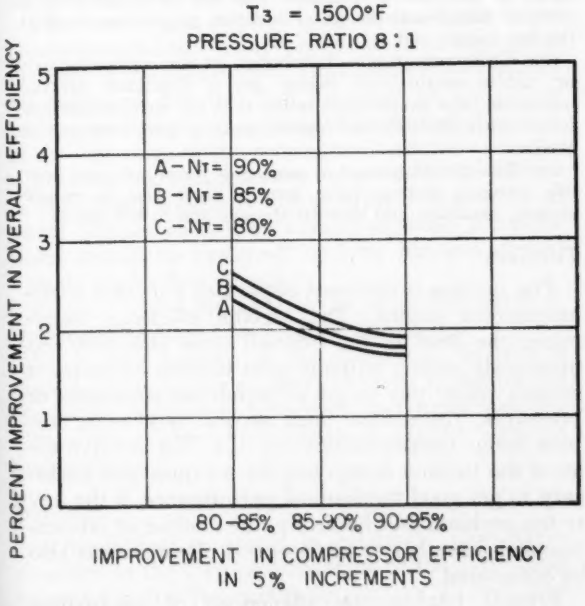

T3 = 1500°F
PRESSURE RATIO 8 : 1
PERCENT IMPROVEMENT IN OVERALL EFFICIENCY
A – NT = 90%
B – NT = 85%
C – NT = 80%
80–85% 85-90% 90-95%
IMPROVEMENT IN COMPRESSOR EFFICIENCY IN 5% INCREMENTS

Figure 8.

efficiency as a function of pressure ratio. At a pressure ratio of 10:1, turbine efficiency is twice as important as at 4:1.

The better the compressor efficiency, the less effect it has on over-all performance. This is shown in Fig. 8. The effect of turbine efficiency as shown in Fig. 9 is independent of the turbine-efficiency range.

The relative ratio of importance of turbine and compressor efficiencies on over-all efficiency is plotted in Fig. 10. This ratio is defined as the per cent improvement in over-all efficiency due to an increment of improvement of turbine efficiency (5 per cent) divided by the improvement in over-all efficiency due to an increment of improvement of compressor efficiency (5 per cent). For the region of high component efficiencies, the importance ratio of turbine and compressor efficiencies on over-all efficiency is greater than at the region of low component efficiencies. The importance ratio decreases at higher pressure ratios. This effect can be applied to part-load operating conditions. For an engine designed for good part-load operation the turbine efficiency is considerably more important than the compressor efficiency at high percentages of full power. At idling and low powers, the compressor efficiency is almost as important as turbine efficiency.

The reason the turbine efficiency is so important is that the work loss due to the turbine inefficiency is so much greater than the work loss due to the compressor inefficiency. This ratio is tabulated in Table 3.

The compressor inefficiency loss is recovered in the combustor in the form of decreased fuel consumption. Fig. 11 is a plot of the percentage of over-all efficiency recovered due to the decreased fuel consumption divided by the percentage of over-all efficincy lost due to the compressor inefficiency.

In summary, good component efficiencies with moderate pressure ratios are preferred to the high-temperature, low-component-efficiency cycle. Good turbine efficiencies are essential to good part-load performance.

If the advantages of engine cooling are to be realized, it will mean going to the complication of a higher pressure-ratio engine or regeneration. Present indications are that cooled turbines can be designed with only a very small compromise in turbine efficiency.

COMPONENT DESIGN CONSIDERATIONS

The reliability of components for a given overhaul life is something that cannot be compromised. Neither can the operating characteristic requirements be compromised below a certain minimum. For those applications where part-load performance is important, it is necessary to have good part-load component efficiencies. To achieve producibility and good mechanical reliability, aerodynamic and thermodynamic compromises are necessary. Both compressor- and turbine-component efficiencies are important; however, the turbine-component efficiency is relatively more important than compressor efficiency and therefore leaves very little room for compromise.

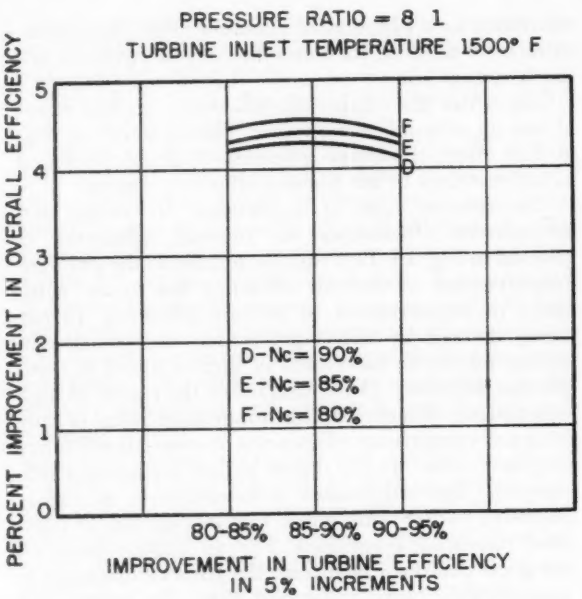


Figure 9.

Compressors

The compressor efficiency is important but can be advantageously compromised. Compressors are sometimes operated at a little higher pressure ratio and a little lower compressor efficiency in order to achieve a higher over-all cycle efficiency.

Aircraft axial designs usually have poor part-load surge characteristics and are therefore unsatisfactory when applied to naval engines.

In matching the turbine to the compressor, the operating line should be properly distant from the surge line. If a compromise in component-design points is necessary to achieve a satisfactory operating line, a compromise in compressor efficiency is preferable to a compromise in turbine efficiency.

For small engines the robustness and high pressure ratios per stage of a centrifugal compressor are advantageous. By compromising a few points of compressor efficiency, a high pressure ratio in a single stage is attainable without all the complications of the many types of stall and surge present in axial compressors. As developments on centrifugal-compressor inducer and diffuser designs proceed, it will probably be possible to achieve good efficiencies at moderate single-stage pressure ratios.

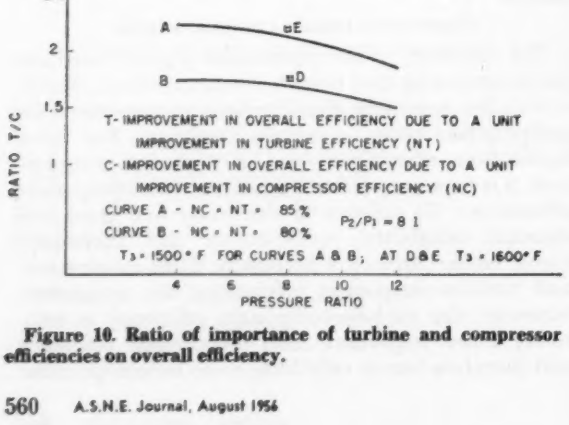

RATIO T/C
A ≅E
B ≅D
T- IMPROVEMENT IN OVERALL EFFICIENCY DUE TO A UNIT IMPROVEMENT IN TURBINE EFFICIENCY (NT)
C- IMPROVEMENT IN OVERALL EFFICIENCY DUE TO A UNIT IMPROVEMENT IN COMPRESSOR EFFICIENCY (NC)
CURVE A - NC = NT = 85%
CURVE B - NC = NT = 80%
P2/P1 = 8 1
T3 = 1500° F FOR CURVES A & B; AT D&E T3 = 1600° F
PRESSURE RATIO

Figure 10. Ratio of importance of turbine and compressor efficiencies on overall efficiency.

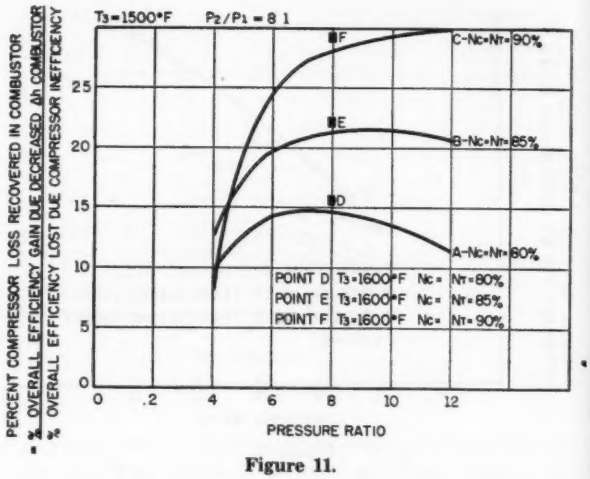

T3=1500°F P2/P1 = 8 1
PERCENT COMPRESSOR LOSS RECOVERED IN COMBUSTOR
= % OVERALL EFFICIENCY GAIN DUE DECREASED Δh COMBUSTOR
= % OVERALL EFFICIENCY LOST DUE COMPRESSOR INEFFICIENCY
C-Nc=Nt=90%
B-Nc=Nt=85%
A-Nc=Nt=80%
POINT D T3=1600°F Nc= Nt=80%
POINT E T3=1600°F Nc= Nt=85%
POINT F T3=1600°F Nc= Nt=90%
PRESSURE RATIO

Figure 11.

Combustion Chambers

Combustion-chamber design is still an art rather than a science. The matching of the dome to the liner and fuel nozzle is a matter of trial and error. The combustor must do the following things sastifactorily:

(a) Good efficiency over a range of operation of mass flows. If part-load efficiency is poor, carbon formation which is harmful to the engines results. Fine fuel atomization over a range of operation is desirable. This can be achieved by a complex fuel-nozzle design or a device to produce swirl at the fuel-nozzle exit.

(b) Good temperature distribution at the combustor exit or turbine-nozzle inlet. Nozzle life is dependent on the maximum inlet temperature rather than the mean or average temperature. Multiple fuel nozzles result in good temperature profiles.

(c) Ease of replacement of combustor liners and good liner life. Ceramic coatings have done a great deal to reduce erosion, oxidation and thermal stresses due to hot spots.

Turbines

The turbine is the most important part of a shaft-horsepower engine. The turbine efficiency determines the level of the over-all cycle efficiency. As previously stated, without good turbine efficiency it doesn't really pay to go to higher temperatures or pressures. There have been several articles on turbine-design considerations (12, 13). The aerodynamics of the turbine design and the compromises necessary to get good mechanical performance is the key to the problem. Mechanically, the method of fabrication and assembly or replacement of parts must also be considered.

Present total-to-total efficiencies of gas-turbine

components are very good; however, diffuser losses reduce the total-to-static efficiencies considerably. It is felt that greater emphasis should be placed on designing efficient diffusers at the turbine exhaust.

Compressor turbines which operate fairly close to a given design point can be designed in accordance with modified free-vortex aircraft-design concepts (14). The power turbine of a multiengine power plant which does not operate at constant u/c values for various pressure ratios should be designed with a little more reaction, more stages, lower relative velocities and good part-load efficiency. For a power turbine operating at almost constant u/c a multistage power turbine is desirable to reduce power-turbine exhaust-diffuser losses.

The trend is toward wider chord blades, higher Reynolds numbers, lower profile losses and greater blade reliability. For power turbines designed for lower relative velocities, a large blade chord will result in flow at a higher Reynolds number and lower profile loss.

For multiengine plants, there is a minimum size of engine for which good axial-turbine component efficiency is obtainable. Because of secondary-loss effects, the turbine-stage efficiency is dependent on the blade height and the tip clearance as per cent of blade height. As blade height decreases to less than 2 in., the stage efficiency decreases. In the power range of less than 500 hp it appears desirable to consider radial turbines.

The potentialities of the radial turbine for small engines will probably be proven in the next few years. Some turbines that have been built and tested have good full- and part-load total-to-total efficiencies. It is merely a question of developing good exhaust diffusers to match the turbines in order to achieve good total-to-static efficiencies.

The duct losses between the power-turbine and compressor-turbine components can be as high as 10 per cent of the total output. The losses can be reduced by compromising the hub diameter of the power turbine so that it is almost equal to the hub diameter of the last stage of the compressor turbine.

Some aircraft turbines are designed for elastic flow in the disks. Weight is not as critical a problem in naval engines as it is in aircraft engines, therefore disks should be designed for long life and should include plastic flow effects.

Power-turbine disks should be of ferritic material. Disk-cooling should be used as necessary.

The use of cooling in the compressor turbine is desirable (for larger engines) in order to reduce the quantities of critical materials and to permit the utilization of lower temperature higher strength materials.

Ducting

Aircraft designs resort to relatively high Mach numbers in the duct sections. One of the first places to reduce engine losses is in the ducting. To keep duct losses to a minimum, low duct velocities between engine components are necessary. For example, in one gas-turbine engine, the sum of the engine pressure losses amount to 23.8 psi. This results in an appreciable loss to the cycle. In addition to designing for low duct Mach numbers, it is desirable to flow-test duct sections (15). A small test program pays big dividends in terms of reduced losses.

Intercoolers

The main advantage of intercoolers is to reduce air-rate requirements. The effect of intercoolers on the cycle is proportional to the compressor inefficiency. For present-day compressors, the complexity of the intercoolers hardly justifies their use. A discussion of the use of intercoolers in compressor units has been published (16).

Regenerators

Regenerator development appears to have reached the point where it is practicable to use a regenerator of 60–70 per cent effectiveness with a continuous-duty or cruising engine of a multibarrel gas-turbine power plant.

The fouling of regenerators is a major problem primarily because of the possibility of steel fires in the cores. The carbon depositing on the gas side sometimes catches on fire and heats up the metal core.

It is necessary to design core sections for minimum fouling and easy disassembly for periodic cleaning.

Evaluation of fouling characteristics and cleaning methods of various core-section designs is presently under way at the U.S. Naval Engineering Experiment Station.

Once the small gas turbine engine's specific fuel consumption is reduced to less than 0.8 lb of fuel per hphr (with an engine pressure ratio of about 5 to 1), a regenerative gas turbine will have practical applications. It is estimated that for a weight increase from 1 to 2 lb per hp, the specific fuel consumption could be reduced from 0.8 to 0.6 lb of fuel per hphr. A gas turbine of 2 lb per hp and 0.6 fuel consumption would be competitive for a great many applications.

Waste-Heat Boilers

The waste heat from the gas-turbine exhaust could be used to supply the heat energy required for hotel-load services; however, it would involve the complication of extra valving tied in with the propulsion plant. The waste-heat boiler would be designed for minimum cruising power on the gas-turbine engines. For a vessel designed to operate at a single speed such as a cargo vessel, the complication is worth the benefit derived from exhaust-heat recovery. For a combatant ship where exhaust heat level and hotel loads are variables the net benefits are questionable; however, a waste-heat boiler would appreciably increase vessel cruising radius.

In any case, since some combatant-type vessels have activity rates of the order of 28 per cent, donkey boilers would be required for hotel-load requirements when the vessel was in port.

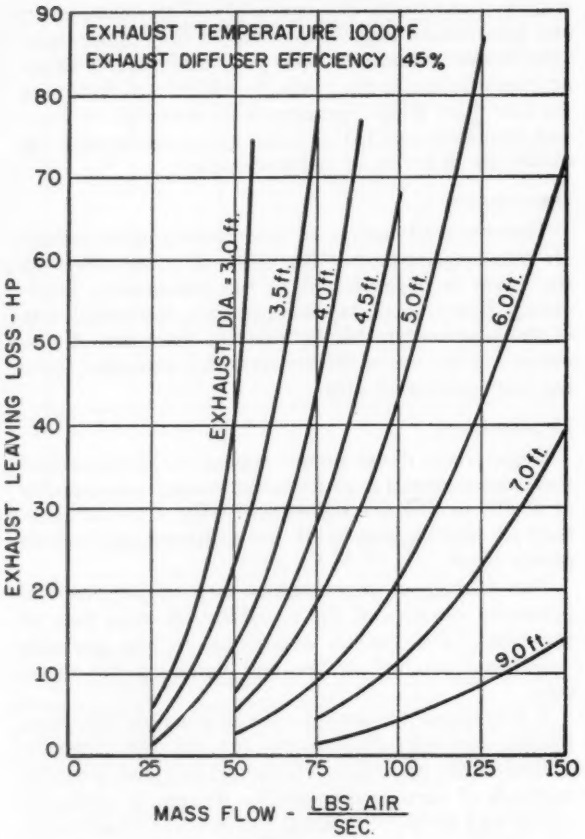


Figure 12.

AIR RATE CONSIDERATIONS

Inlet- and exhaust-duct sizes for an engine installation are dependent on the engine air rate, exhaust-gas density and the allowable exhaust stack velocity. The high air rates of gas turbines have been one of the problems associated with ship-installation studies. Ducts are sized for the full-load air rate. For duty profiles where the percentage of time at full power is small it is desirable to take the power loss associated with going to higher exhaust velocities. Suitable silencing methods should be provided at the stack exhaust to reduce full-power noise level (17).

Figs. 4 and 5 illustrate the optimum thermodynamic efficiency and air-rate design points for various regimes. The optimum efficiency design point can be compromised with the optimum air-rate curve. The duty profile should provide the necessary weighting factor.

Figs. 12-15 illustrate the relative importance of exhaust-stack frictional losses and velocity leaving losses. Figs. 12 and 13 are based on some recent friction data that may be a little on the low side; however, the frictional losses are less than the leaving losses even for more conservative friction data.

It is concluded that for straight exhaust stacks it is desirable to use higher stack velocities (about 300 fps) and recover the velocity head at the top of the exhaust stack with a diffuser. Where the exhaust duct is not straight or where deck openings and duct sizes are not required to be kept to a minimum, lower exhaust velocities should be used. Because of the importance of back pressure on gas turbines the aerodynamics of the exhaust duct must be considered.

CONCLUSIONS

In considering duty-profile requirements for naval-compatant vessels, the main advantage of the gas turbine as in the case of emergency and booster engines is its low specific weight. It remains to be determined how much of this weight saving can be used for increased fuel tankage or other equipments.

The over-all reliability of an engine is only as good as the product of reliability factors for engine components and accessories in series. Components and accessories should be kept to the minimum necessary for pushbutton operation with good part-load fuel consumption.

Good maintenance is dependent on the concept of excellent reliability within the operating overhaul period, as well as designing for rapid disassembly and replacement of complete component sections (such as turbine rotors). Down time can be kept to a minimum, resulting in increased availability of the ship and decreased maintenance personnel require-

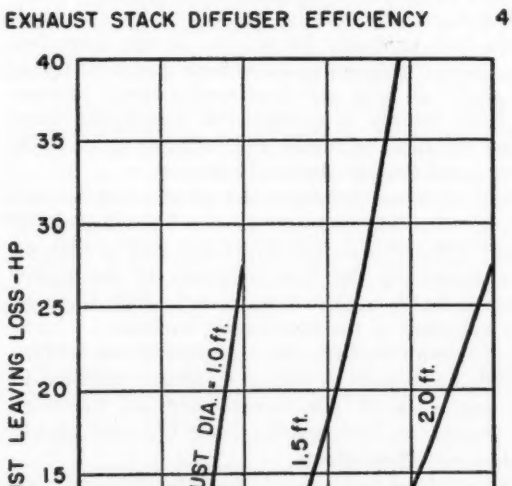


Figure 13.

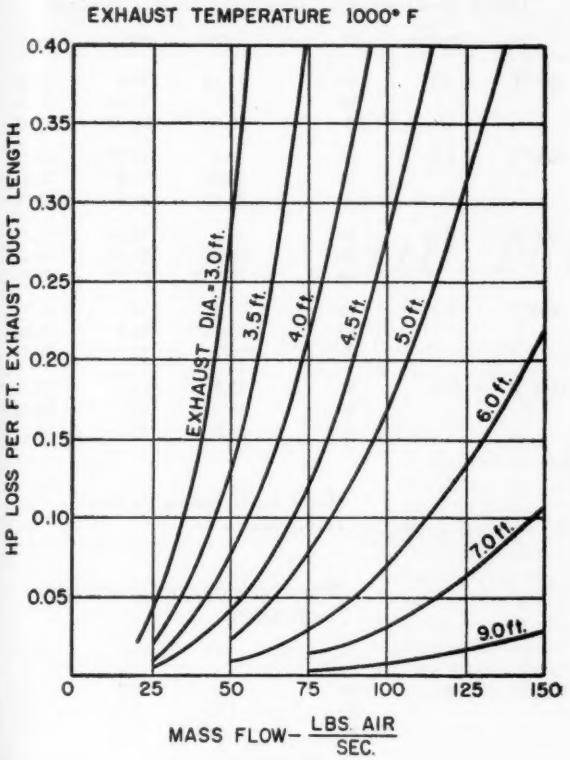


Figure 14.

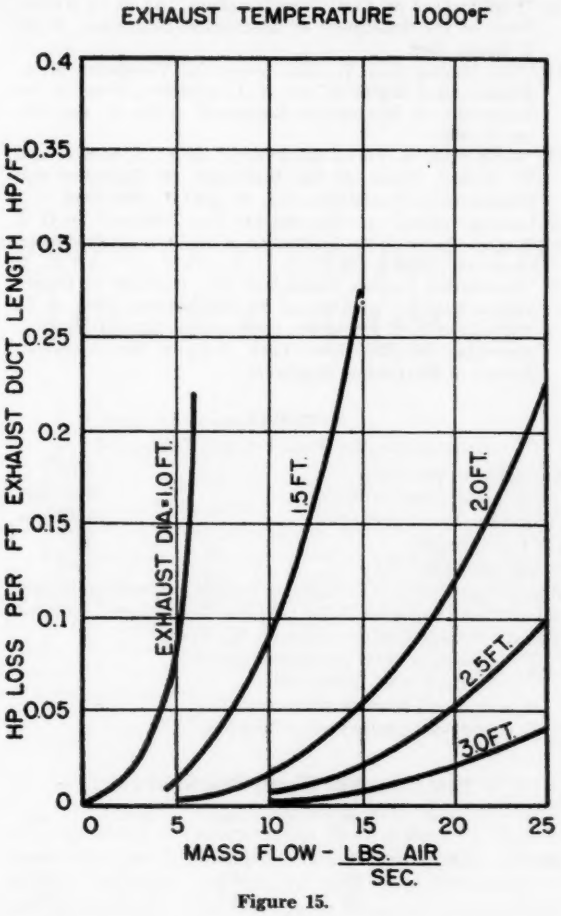


Figure 15.

ments. By replacing complete sections and shipping small sections to a repair base for overhaul the level of training of ship's service personnel will be a bare minimum. Small sections could be stored on board and periodically deposited at a repair base. Auxiliary engines designed for 1000 hours between overhauls would require more overhauls than other engines; however, the ease of overhaul, small size of parts and the small number of parts subject to failure should make up for this. It is believed possible to obtain better than 1000 hours at full power between overhauls at some time in the future.

The duty profile or required operating characteristics determines the thermodynamic regimes for which a power plant is designed. The compromises in compressor efficiency to obtain pressure ratio or the compromises in turbine efficiency to obtain higher turbine-inlet temperatures are dependent on the duty profile and thermodynamic considerations. On paper, gas-turbine engines look as if they can be designed for maximum simplicity and reliability. The proof of the paper studies can only be determined by obtaining fleet operating experience.

BIBLIOGRAPHY

1. "The Gas Turbine as a Prime Mover on U. S. Naval Ships," by Cdr. W. M. M. Fowden, Jr., USN, R. R. Peterson, and J. W. Sawyer, ASME Paper No. 53-A-233.
2. "British Naval Gas Turbines," by G. F. A. Trewby, Trans. ASME, vol. 77, 1955, pp. 561-583.
3. "Shell and the Gas Turbine," (no author), Gas and Oil Power, March, 1954.
4. "Operating Experience on General Electric Gas Turbines," by W. M. Moyer, ASME Paper No. 54-A-160.
5. "Gas Turbines for Ship Propulsion," by John McMullen, presented June 6, 1955, to the 27th Annual OGP Conference, Washington, D.C.
6. "Evaluation of a Naval Gas Turbine," by J. A. Culver and A. L. Wood, ASME Paper No. 55-S-44.
7. "Prospects of Gas Turbines in Naval Applications," by R. T. Simpson and W. T. Sawyer, ASME Spring Meeting Paper, 1950.
8. "Recent Naval Steam-Plant Design," by Cdr. C. H. Meigs, USN, Trans. SNAME, vol. 62, 1954, pp. 242-271.
9. "General Considerations in Naval Shipboard Power Machinery Development," by I. C. Jackson, Jr., *Journal of the American Society of Naval Engineers*, vol. 65, no. 2, May, 1953.
10. "A Comparison of the Total Weight and Bulk of In-Line Naval Gas Turbine Engines of Differing Degrees of Complexity," by D. F. Collins and D. W. Thomas, Trans. of the N.E. Coast Institution of Engineers and Shipbuilders, vol. 71, part 6, April, 1955.
11. "Machinery of the Immediate Future," by J. E. Church, abstract of Institution of Marine Engineers paper appeared in the Shipping World, April 20, 1955.
12. "Some General Considerations in the Design of Axial Flow Gas Turbines," by Sumner Alpert, address presented to the SAE, San Diego Section, September 11, 1951.

13. "Performance of Axial Flow Turbines," by D. G. Ainley, Proc. of the Institution of Mechanical Engineers, W. E. P. No. 41, 1949.
14. "The Marine Gas Turbine From the Viewpoint of an Aeronautical Engineer," by A. H. Fletcher, Proc. of the Institution of Mechanical Engineers, series A, vol. 166, no. 2, 1952.
15. "Fluid Flow in Turbo Machinery," by H. S. Fowler and V. Walker, Trans. of the Institution of Engineers and Shipbuilders in Scotland, vol. 97, part 3, 1953-1954.
16. Correspondence on "The Marine Gas Turbine," by G. I. Pauley, Trans. of the Institution of Mechanical Engineers, December, 1953, p. 197.
17. "Installation Factors Related to Gas Turbines in Closed Compartments," by Cdr. L. W. Shallenberg, USN, J. O. White, and K. E. Schlacter, presented at Annual Meeting, November 30, 1952, New York, N.Y., of The American Society of Mechanical Engineers.

APPENDIX

Assumptions for Figs. 4-9 and Tables 1-3

Ambient temperature ..... 80 F
Compressor Inlet pressure ..... 14.55 psia
Turbine exhaust pressure ..... 14.92 psia
$\Delta P/P$ combustor 5%
Combustor Efficiency 98%
Mass of gas due to fuel addition equals air leakage losses.
Calculations based on "Gas Turbine Gas Charts." Bureau of Ships Research Memorandum No. 6-44.
P_2/P_1—pressure ratio across compressor
T_3—turbine inlet temperature
N_t—over-all turbine efficiency
N_c—over-all compressor efficiency

TABLE 1—*Net Work, Btu/lb of air*

T_3	P_2/P_1	N_T	N_C	80%	85%	90%
1500°F	4:1	80		41.4	46	50.2
		85		48.9	53.5	57.7
		90		56.4	61.0	65.2
1500°F	6:1	80		40.3	46.8	52.3
		85		49.8	56.3	61.8
		90		59.1	65.6	71.1
1500°F	8:1	80		39.5	47.2	54.0
		85		50.2	57.9	64.7
		90		61.0	68.7	75.5
1600°F	8:1	80		46.3	54.0	60.8
		85		57.5	65.2	72.0
		90		68.5	76.2	83.0
1500°F	12:1	80		26.2	36.0	44.8
		85		38.3	48.1	56.9
		90		50.3	60.1	68.9

TABLE 2—*Over-All Cycle Efficiency, Per Cent*

T_3	P_2/P_1	N_T	N_C	80%	85%	90%
1500°F	4:1	80		14.15	15.6	16.65
		85		16.7	18.05	19.15
		90		19.3	20.55	21.65
1500°F	6:1	80		15.4	17.97	19.1
		85		19.05	21.0	22.6
		90		22.6	24.5	25.98
1500°F	8:1	80		16.5	19.15	21.3
		85		20.95	23.45	25.5
		90		25.5	27.83	29.8
1600°F	8:1	80		17.3	19.7	21.7
		85		21.6	23.8	25.65
		90		25.7	27.8	29.6
1500°F	12:1	80		12.97	17.0	20.3
		85		18.97	22.7	25.75
		90		24.9	28.3	31.2

TABLE 3—*Ratio Work Loss due to Turbine Inefficiency / Work Loss due to Compressor Inefficiency*

T_3	P_2/P_1	N_T	N_C	80%	85%	90%
1500°F	4:1	80		1.91	2.7	4.35
		85		1.43	2.03	3.26
		90		.955	1.35	2.18
1500°F	6:1	80		1.72	2.45	3.83
		85		1.28	1.83	2.86
		90		.86	1.22	1.91
1500°F	8:1	80		1.63	2.3	3.65
		85		1.22	1.73	2.74
		90		.81	1.14	1.81
1600°F	8:1	80		1.69	2.4	3.79
		85		1.27	1.8	2.84
		90		.839	1.2	1.90
1500°F	12:1	80		1.45	2.05	3.27
		85		1.08	1.53	2.45
		90		.725	1.025	1.63

ROBERT L. KELLY

THE PRELIMINARY DESIGN OF SURFACE WARSHIPS

THE AUTHOR

graduated in 1946 from Webb Institute of Naval Architecture. He served in the Navy until discharged at which time he was employed by W. C. Nickum & Sons in Seattle, Washington. In 1948 he joined the staff of the Scientific and Test Section of the Puget Sound Naval Shipyard where he remained until 1950 when he reported to U. S. Coast Guard Headquarters to work in the Naval Engineering Section. Shortly after, he joined the Preliminary Design Section of the Bureau of Ships where he remained until 1955 when he became Program Planner of Research and Development in Naval Architecture for the Bureau of Ships. Mr. Kelly was Secretary-Treasurer of the Chesapeake Section of the Society of Naval Architects and Marine Engineers from 1953–1955 and is presently Vice-Chairman of the group.

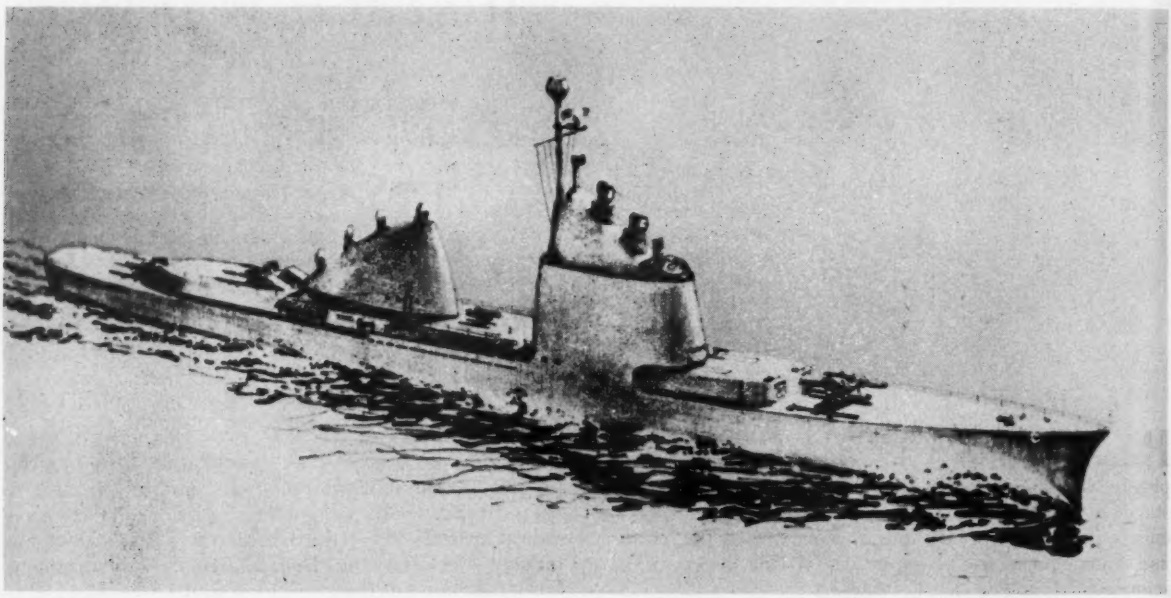
PRELIMINARY design is one of the most interesting phases of naval architecture. It begins in a dream stage, and because all the naval architect's experience, originality, and imagination are called into use, may be more properly called an art than a science.

The preliminary design of a surface warship is usually initiated by a request from the Ship Characteristics Board, a part of the Office of the Chief of Naval Operations. Because of the inadequacy of existing ships, technological advances, or new concepts of warfare, the Ship Characteristics Board may request a design similar to existing ships or a totally new type of design. This request from the Board is the naval architect's starting gun and the beginning of a new adventure for him. He may be supplied only with a vague concept of the ship's mission and desired speed. From this information he begins an exploratory series of design studies, appropriately called "Spring Styles." These consist of an inboard profile and a bird's eye view of the design's topsides. Through these "Spring Styles" he graphically transforms his thoughts into a series of ship designs. From these designs he is able to make some broad brush weight estimates, assign vertical centers to the weights, check stability, calculate the power required to make the desired speed, and work out a variety of topside arrangements. These are then presented to the Ship Characteristics Board to be used as a guide in drawing up the preliminary set of characteristics. This set of characteristics will define the design's mission, speed, endurance, type of propulsion, armament, complement, and displacement that the Ship Characteristics Board desires. These characteristics are tentative and a great deal of interplay between the Board and the Preliminary Design Branch of the Bureau of Ships will take place before the final characteristics are completed. Because so many forces are at work on the Ship Characteristics Board, the preliminary characteristics may call for features that are incompatible with one another. Too many guns, too high a speed, or too many secondary missions are often the cause of this incompatibility. These must be resolved by the Board and the naval architect. This forms the interplay of design and develops in the naval architect the preliminary design attitude that nothing is certain; nothing is final. This interplay continues throughout the development of the design even though work progresses far beyond the "Spring Style" stage. In some cases design studies number up to fifty and in one case they totalled seventy-seven. This is the atmosphere of preliminary design. Out of all this process, the Ship Characteristics Board is able to weigh the desirability of certain design features against their cost and reach a compromise design that they define in a final set of characteristics.

Between the "Spring Styles" and the final characteristics the naval architect is involved in an intensive design process. All estimates made in the "Spring Style" stage are recalculated in greater de-

tail. Longitudinal centers are given to the weights to find the condition of trim. A lines plan is drawn of the hull, and a plan is made of the hull appendages. These are sent to the model basin where a model is made of the design and tested. Curves of statical stability are calculated. An arrangement plan is made in more complete detail, and damage stability calculations are performed. A midship section is drawn and the design's scantlings are adjusted to insure adequate hull strength. Detailed structural calculations are also started and usually continue until the end of the preliminary design. When a balanced design fulfilling as many of the preliminary characteristics as possible is completed, its features are reported to the Ship Characteristics Board. From this report the Board draws up the final characteristics of the design.

Following this, a wrapping up of the design begins. Structural plans are completed with their corresponding weight estimates. A final arrangement plan is finished, and a review is made of all calculations and considerations of the design. If this review shows that a balanced design has been achieved, the preliminary design is considered completed.



Official United States Navy Photograph

Newest of the New Navy—The "Salu"-type nuclear-powered, guided missile cruiser designed to be equally effective against sea, air, land, and undersea targets. This artist's conception portrays the ship, subject to design changes, which has been requested as a part of the 1957 construction program. The latest addition to the Navy's atomic attack and defense systems will carry a number of multiple missile launchers for missiles of varying range and capabilities plus torpedo launching equipment.

II

Certain physical features of a ship such as length, beam, depth of hull, freeboard, and weight make up the main elements of design. These features largely define the ship, and a description of how they are selected follows.

The length of a new design is determined in two stages: the first by an estimate, and the second through a reasoned solution. The estimate is made by comparing the design's functions and type of machinery with those of an existing ship. The design is lengthened or shortened according to its similarities with, or its differences from the existing ship. This estimate is typical of the initial stage.

One exception is the aircraft carrier which is essentially a floating airport. Here, the length of airplane landing and launching areas pretty well determine the design's length from the beginning.

For other ships the reasoned solution usually determines the design's length. The reasoned solution is reached by drawing an arrangement plan of the ship's interior. The arrangement develops throughout the intermediate stage of design from a simple allocation of spaces within the ship to a fairly detailed drawing of the spaces. When all the functions are included within the hull, and gun and topside equipment clearances are adequate, in ships other than carriers, the arrangement is considered final. Because this results in the shortest ship possible, which is also the most inexpensive ship, this finally determines the design's length.

The beam of a ship is governed by three considerations. These are stability, the capability of fitting into existing docking facilities, and in the case of landing craft, giving the ship adequate bouyancy to meet minimum draft requirements. The metacentric height of warships is usually expressed as a fraction of the ship's beam. The size of this fraction depends upon the type of ship and the severity of the damage the ship may be expected to experience. Once the

order of this fraction has been established for a type of ship, it becomes a handy tool for correcting the design's beam following a weight estimate. The final dimension of the beam usually depends upon the outcome of the damage stability calculation, although in some cases of designs with low metacentric height the condition of wind heel dominates. In most cases efforts are made to eliminate off center flooding allowing the beam to be reduced and the draft to increase. This increase in draft lends the design superior capabilities in rough weather.

The depth of a new design is determined by five factors. These factors are strength, draft, freeboard, low silhoutte, and the vertical dimension of the propulsion machinery. Generally all elements will be accommodated by using a normal depth to length relationship.

Much thought is given to freeboard. Increased freeboard is one practical answer to the problem of seaworthiness. Tabulations of past and present ship's freeboard to length ratios at pertinent hull stations can be a helpful guide for new designs. One of the most interesting freeboard criteria is that of finding the moment about midships of the reserve bouyancy in the ship's forward quarter. When this moment is divided by the product of the ship's longitudinal gyradius and its displacement, it yields a coefficient, which plotted against the ship's length shows excellent correlation with reports of ship's dryness.

A large part of the naval architect's time is spent estimating the weight of a new design. Estimating is simplified when weight estimates of similar existing ships are available. If they are available, weights of the new design can be proportioned by comparing its dimensions and coefficients with those of the existing ship. This greatly assists in preparing weights for the initial and intermediate stages of preliminary design. The weights of similar ships are also valuable checks for the final weight estimate of a design.

The specific factors used for proportioning weights are best developed by the individual naval architect. A splendid set of factors is used by the Navy Department. They are based on principal dimensions, displacement, speed, power, hull coefficients, and their products.

Margins of weights are also important in ship design. In the preliminary design stage they should cover two developments. The first is an increase of displacement between the estimated weight of the preliminary design and the weight of the finished ship. The second is the increase in weight that the ship will experience in its lifetime. These margins are determined by analysis of past ship's weights and a guess at the future, although tedious and slow, weight estimating is the very heart of a ship design. With the conclusion of weight estimating this discussion will pass on to factors of secondary importance in preliminary design.

Although some elements are defined as primary and some as secondary, the division is arbitrary and all must be given full consideration in each of the three stages of preliminary design.

The stability of a ship is largely determined by two types of calculations. The most important is the damage stability calculation. This is made by using a quick method in the intermediate stage, and by a more detailed calculation employing an integrator in the final stage of preliminary design. The other calculation is that of statical stability. When statical stability studies have been completed, the last freeboard adjustments are made, and the design's stability is usually found to be adequate because the damage stability criterion is often more severe than that of the initial stability for an intact ship.

Only a few words need to be said about hull coefficients. Conventional naval ships travel at two important speeds. These speeds are specified by the characteristics. They are the trial speed and the speed at which a ship should travel a certain number of miles. The latter speed is called the ship's endurance speed and is considerably lower than the trial speed. Since a greater part of the ship's life is spent traveling at the endurance speed, hull coefficients favoring this speed would reduce the amount of fuel necessary for the design. However, coefficients that reduce the resistance of the design at the lower speed increase the resistance at the higher speed. This often requires an outsized machinery installation. So a compromise in coefficients is reached by balancing weight and cost of machinery against that of oil.

The power and fuel oil necessary to make the required speed and endurance are important goals to be reached by the naval architect. The power for a design of normal form is estimated by using Taylor's Standard Series. It is best to use margins in calculating the power and amount of fuel oil. Again these margins may be calculated from analysis of past designs.

The last element of preliminary design is that of ship's strength. In preliminary design, strength calculations are made to check hull scantlings and to assist in making weight estimates. The main hull stresses are found by calculating the strength of the midship section. In the early stages a constant is used to find the bending moment. Later, the bending moment is calculated from the ship's loading. Special strength problems are investigated fully in the preliminary design stage. Finally the ship's shell and decks are checked for hydrostatic and local loading and for buckling in compression. Complete and detailed strength calculations are left to be made by others at later stage of the design.

This concludes the description of the preliminary design of surface warships, but not the story of the design itself. There is always that day when the naval architect sees his design in the form of a ship sliding down the launching ways or taking its place in the fleet. Then a whole new story begins in which his design plays its part. This is the story of courage and valour that makes the tradition of the United States Navy.

33,000 HOURS ON MARINE GAS TURBINES IN NAVAL SERVICE

ACKNOWLEDGEMENT

This is a slightly abridged version of a paper presented at the American Society of Mechanical Engineers' Gas Turbine Power Conference in Washington, D. C., 16-17 April, 1956. The authors are Commander W. M. M. Fowden, Jr., USN, and J. W. Sawyer, of the Gas Turbine Branch of the Bureau of Ships.

INTRODUCTION

THE MAJOR advantages of gas-turbine engines have been forcefully demonstrated many times in recent years. Some of the proven strong points of this relatively new prime mover include the following:

1. Light weight. (1)[1]
2. Reliability.
3. Low maintenance.
4. Small space (1).
5. Rapid starting.
6. Low installation cost.
7. Easy starting in cold weather.
8. Few moving parts.
9. Few auxiliary units needed.
10. Low cooling water requirements.
11. Small quantity of repair parts.
12. Economy.

Industrial applications of gas turbines for prime movers in electric power generation (2), transportation and oil and gas pumping stations are growing rapidly. The oil and gas industry has taken some major forward steps in this respect. One outstanding example is the Lake Maracaibo, Venezuela, installation where ten 6000-hp units have been installed. These units have now logged 73,000 operating hours (3). There are twelve 8000-hp units being added at this site. When these new turbines are installed there will be a total of 156,000 hp in gas-turbine engines at this location. It will be the greatest concentration of gas-turbine power in the world. It is of particular interest to note that one gas-turbine-driven pumping unit in the United States has completed 30,000 hours of operation (4).

The Union Pacific Railroad has had 25 gas-turbine-driven locomotives in operation for several years. These units have logged 4,750,000 operating miles. This road recently ordered 45 new gas-turbine-driven locomotives. Each locomotive will deliver 8500 hp at an ambient temperature of 100°F., which is as much power as can be delivered by five conventional diesel-electric locomotives (5). The additional power will be used to accelerate freight-train speed rather than increase train tonnage.

Marine applications of high-horsepower gas turbines are now being made in England and in the United States. The tanker *Auris* is having all diesels removed and a 5500-hp gas-turbine propulsion plant installed as a replacement (6). The U.S. Maritime Administration is installing a 6000-shp marine gas turbine in a Liberty Ship, *John Sergeant* (7). Eight new merchant vessels of the Blue Funnel Line are now being equipped with 350-kw gas-turbine-driven emergency generator sets for standby and emergency use (8).

The Corps of Engineers has a 5000 kw gas-turbine-driven generator mounted on two railway cars. This unit is designed to provide power for troop support and also for use in disaster areas (2).

The gas turbine is also finding a firm economic niche in commercial aviation. The turboprop engine recently placed in commercial service in the U.S. is proving attractive, while the Viscount aircraft used in commercial service throughout the world, recently logged 90,000 hr (9).

As noted above one reason for the wide applications of gas turbines is economy. The following maintenance-cost comparisons of gas turbines versus other prime movers points this out (10):

[1]Numbers in parentheses refer to the Bibliography at the end of the paper.

1. Tanker *Auris* gas-turbine propulsion 0.06 cent per kw hr.
2. Brown Boveri gas turbines (based on service life of 1000 hr.) 0.023 cent per kw hr.
3. General Electric Bell Isle gas turbine 0.004 cent per kw hr.
4. Diesels (British Diesel Engine Users Association) 0.30 cent per kw hr.
5. Steam turbines (without boilers—British Electricity Authority) 0.049 cent per kw hr.

U. S. NAVAL APPLICATIONS

Naval applications of marine gas turbines have increased rapidly during the past few years. Installations of gas turbine engines which have been made and are now in use include emergency generators, propulsion and auxiliary generators. At the present time a total of 55 vessels are in operation with gas-turbine engines installed. These include a destroyer, landing craft and a number of minesweepers as shown in Table 1. The number of engines installed includes 200 individual units. The total installed horsepower as of 1 November 1955 was 32,564. Operating time on the installed engines up to 1 November 1955 was 33,327 hours. From Table 1 it is apparent that the great majority of the operating hours was built up on engines driving auxiliary generators.

TABLE 1

SUMMARY OF MARINE GAS TURBINE ENGINES NOW USED ON U. S. NAVAL VESSELS

Engine Designation		Rating		Vessel		Number Engines Installed	Total HP Installed	Total Hours Operation[1]
Navy	Mfgr	HP	Application	Type	Number			
4MC205-1	Solar T-520	500	Emergency generator	Destroyer	1	1	500	103
4MC205-1	Solar T-520	500	Auxiliary generator	Minesweep	1	2	1,000	1,000
2MV27B-6	Boeing 502-6	160	Auxiliary generator	Minesweep	46	184	29,440	31,231
2MV27B-8	Boeing 502-8C	160	Auxiliary generator	Minesweep	2	2	320	30
2MV27B-2	Boeing 502-2	160	Propulsion	Landing Craft	1	1	160	180
2MV27B-8	Boeing 502-8C	160	Propulsion	Minesweep	4	4	640	564
1MC4S-1	Solar T-45	46	Pump Drive	—	—	2	92	209[2]
1MCS-1	Solar T-45	46	Deicer	—	—	2	92	5[3]
2MCB-6	Boeing 502-6M	160	Deicer	—	—	2	320	5[3]
			Total		55	200	32,564	33,327

[1] Up to 1 November 1955.
[2] Seven units procured—Two in fleet for operational evaluation
[3] Two units in fleet for operational evaluation

ENGINE REPAIRS

As the operating personnel have gained experience on gas-turbine engines, primarily those installed aboard the MSB-5 Class minesweepers, they have been making minor overhauls of the engine and replacing rotating parts and auxiliary equipment. It is conceivable that after some additional experience, complete engine overhauls can be made by the Fleet provided a sufficient number of engines are in operation to warrant setting up the necessary repair facilities and equipment. At the present time engines from the MSB-5 Class minesweepers are being sent back to the manufacturer for complete overhaul at the end of 500 hours of operation. The overhaul time will probably be increased to 1000 hours in the next few months. Another engine in the same class is on a 1000-hour overhaul schedule.

It is of interest to note that procurements for new building programs only include 25 per cent complete spare engines; i.e., if a shipbuilding program were initiated that would require 100 gas-turbine engines of a particular type only 25 complete engines would be ordered as spares. This is a considerable reduction in the number of spare engines which would be ordered for other types of prime movers.

250-KW EMERGENCY GENERATOR SET

A 500-hp gas-turbine emergency generator set is in operation in USS *Timmerman*. The rated output of the set is 250 kw. The set supplies emergency power in the event of failure of both normal and alternate ship's service electrical systems. The gas-turbine-driven generator is started automatically, upon failure of the above systems, and will carry full load within 30 seconds.

A 400-hp gas turbine, the Solar T-400, was a part of this emergency generator set (11) when it was first installed. In 1954 this engine was replaced with a 500-hp gas turbine engine, the 4MC205-1 (12) (Solar T-520) (13). Data on the engine are given in Table 2.

The turbine power plant consists of five assembled components and the automatic start and control system. These five components are as follows: Main reduction gear and accessory drive, compressor, turbine, exhaust collector and the single through-flow combustion chamber. The gear housing, compressor turbine and exhaust collector are bolted together in line. A splined coupling shaft provides a direct drive through the gearbox and at a reduced speed to the alternator. Engine accessory components consist of fuel pump, lubricating-oil pump, governor, speed-sensitive control switch and tachometer generator, mounted on and driven by the reduction-gear assembly. Automatic starting is accomplished through an electric motor mounted on the gear box. A manual switch is provided on the control panel to shut down the unit. The operation is closely controlled by a series of actuating relays in the turbine control box and alternator control panel. Ignition and the fuel flow required at the increasing turbine speeds during the starting and at varying alternating loads are controlled automatically. Automatic safety shutdown devices protect the engine against excessive exhaust temperature, overspeed, low oil mist pressure, and high oil temperature. Manual controls are incorporated which can be used to override the automatic features of the system in case of emergency.

The turbine and generator set aboard *Timmerman* has been operated a total of 103 hours as of 1 November 1955. The operation has been generally satisfactory. There has been no trouble with the turbine, the compressor rotating elements or with the combustion-chamber liner. Some minor troubles have occurred with engine auxiliaries and components. However, these are considered to be of a very minor nature and they have been or are being corrected so that no future operational difficulties are expected. Inspections were made of the turbine at the end of 50 hours of operation and again at the end of 100 hours. The routine inspection following the 50-hour run revealed the following:

1. The igniter plug was dirty but not damaged.
2. A very slight carbon build-up on the combustor liner was noted. Apparently small chunks have been carried through the turbine but with no damage to the engine.
3. The combustor liner was removed from the can and inspected. No damaged areas were noted.
4. The inlet to the turbine was in good condition.
5. Fuel nozzle was removed and cleaned. (soft coating on the nozzle.)
6. The primary fuel filter was replaced.
7. The lube-oil filter and generator-bearing oil-mist air strainer were inspected.

During the 100-hour inspection it was noted that the inducer section and the compressor were coated with a mixture of oil and some hard foreign material. The inducer vanes were cleaned by hand and then the compressor was cleaned by injecting a quantity of soft abrasive cleaning compound into the compressor while operating at 14,000 rpm. Prior to this operation, the engine had been experiencing a series of shutdowns due to high engine-exhaust temperatures. It was believed that the build-up of foreign material on the compressor had restricted the air flow so that high exhaust temperatures resulted. After the cleaning of the compressor the engine was checked and found to operate satisfactorily. Before cleaning the turbine-exhaust temperature had been 985°F, with a load of 200 kw and a turbine-inlet temperature of 1395°F. After cleaning, the exhaust temperature was 910°F, with a load of 200 kw and a turbine-inlet temperature of 1295°F. The combustor and the compressor were inspected and found to be in excellent condition after this treatment.

The emergency generator set on this vessel was not placed in the most advantageous position to suit the gas-turbine engine. The unit was located far below the main deck and extensive runs of ducting were necessary. Some of the operational problems experienced have been directly associated with the long runs of inlet and exhaust duct. During a recent voyage the ship encountered rough weather and a considerable amount of green water was brought aboard and entered the turbine. Following this experience an unsuccessful attempt was made to start the unit. An investigation showed that salt water had carried down ducts and salt crystals had formed around the compressor-blade tips to the extent that the starting motor was not able to turn the shaft. A torque was applied to the shaft by hand and it was broken free with a torque of about 60 in-lb. After break-away the unit was started with no trouble.

This experience has pointed out the importance of providing inlet and exhaust ducting with the most practicable protection against salt-water entry, and also with adequate means for removal of salt water in the event it does enter the duct. A rather large plenum chamber is desirable at the bottom of the inlet ducting in order to prevent salt water from entering the engine. This plenum chamber also must be provided with a suitable drain line. Similar protection is likewise necessary on the exhaust ducting. Studies are currently under way to correct and prevent recurrence of this trouble. In general, the operation of this 500-hp gas turbine as an emergency generator drive aboard *Timmerman* has been satisfactory. The minor troubles that have been encountered with auxiliaries are not disturbing and it is believed that they have now been overcome by a redesign and replacement with improved parts.

500-HP AUXILIARY GENERATOR DRIVE

Two 500-hp gas turbines each driving an auxiliary generator have been installed in a vessel of the MSB-5 Class. The gas turbine is the same model as that used

TABLE 2

DATA ON SOLAR T-520 GAS TURBINE

HP	500
Description	Single shaft
Mfgr Designation	T-520
BUSHIPS Designation	4 MC205-1
Weight	450 lbs.
SFC full power	.95
Length	60″
Width	33″
Height	34″
Air flow	8.82 lbs./sec.
Fuel	MIL-F-16884A (Diesel oil)
Pressure Ratio	4.57
Turbine inlet temp.	1465°F.
Application	Generator Drive

at the present time to power the *Timmerman* emergency generator set. Engine data are presented in Table 2.

The engine-generator combination is bolted to a rigid sub base in the vessel which permits ease of installation and removal. The sub base is secured to seating assemblies in the compartment.

The turbine assembly consists of a three-stage axial-flow turbine, a stator and nozzle assembly, inlet scroll and exhaust collector. The rotor and stator are both overhung and supported on the cast-aluminum bearing-support housing which is attached to the after end of the compressor. The 354-blade turbine rotor disk and shaft rotate in a matched set of ball thrust bearings and a single set of roller bearings.

The two gas-turbine-driven generator units on this minesweeper have been operated for a total of 1000 hours as of 1 November 1955. As in the case of the 250-kw emergency generator just described, some minor troubles have been experienced with auxiliaries. However, there have been no major troubles with the compressor, the combustor or the turbine. Some of the problems encountered are as follows:

1. Low fuel-oil pressure.
2. False start due to low battery voltage.
3. Damaged O-ring in governor and fuel pump.
4. Leak in lube-oil manifold and scavenge pump.
5. Broken coupling between splines on lube-oil and scavenge pump. Replaced second-stage turbine cooling-air shaft carbon seal.
6. Replaced combustor liner because of cracks and distortion.
7. Refitted oil breather line to exhaust collector due to broken fitting on exhaust shrouding.
8. Cleaned igniter plug.
9. Replaced primary and secondary fuel filters.
10. Replaced second-stage air shaft carbon seal.
11. Installed new turbine-exhaust pyrometer and fuel pressure gauges.

At the 500-hour inspection the following action was taken:

1. The cooling-air seal was renewed.
2. The combustor liner was renewed.
3. The igniter was cleaned.
4. The engine and compressor were cleaned.

It was noted during inspection that a considerable build-up of oil and foreign material had occurred on the compressor blades and inducer. The abrasive material mentioned earlier was injected into the air inlet and did a satisfactory job of cleaning the unit. Operations to date have been good in so far as the turbine itself is concerned. The next major inspection is scheduled for 1000 hours after which an additional 500 hours will be run for a total of 1500 hours.

160-HP MINESWEEPER GENERATOR DRIVE

The Boeing Model 502-6 engine, Table 3, is the one used to power the auxiliary generator sets on the MSB-5 Class vessels. Forty six of these vessels have been equipped with this engine and a total of 184 units have been installed. The engine is unique in that it is the first marine gas-turbine engine to be built in production quantities. A number of marine gas-turbine engines of other types have been built and installed. However, none of these has been built in quantity production.

The basic engine is rated as a 160-hp unit. These engines are used in pairs to drive the auxiliary generator, and two generator sets are installed on each vessel. Each engine feeds into a combining reduction gear which in turn drives the generator at a normal speed of 2000 rpm. The input pinion from each of the gas turbines is attached to a flexible over-running clutch. The compressor on this engine is a single-stage centrifugal type with the normal pressure ratio of 3:1, with 3.5 lb. per sec air-mass flow. The first-stage turbine is direct-connected to the compressor to form the gas-producer section. There are two combustion chambers connected by a cross-fire tube. The first-stage rotor turns at 36,500 rpm normal maximum and the second-stage turbine is a free-wheeling unit which turns at 23,725 rpm normal. The engine has a

TABLE 3

DATA ON BOEING 502-6 GAS TURBINE

HP	160 HP max continuous
Description	Split shaft turbine
Mfgr Designation	502-6
BUSHIPS Designation	2 MV 27B-6
Weight	252 lbs.
SFC full power	1.4
Length	42.90″
Width	23.20″
Height	23.34″
Air flow	3.6 lbs./sec.
Fuel	MIL-F-16884A (Diesel oil)
Pressure Ratio	3.1
Turbine inlet temp.	1575°F.
Application	Generator drive Deicer (Modified version of 502-6)

built-in reduction gear with a ratio of 8.62:1. Accessories on the engine in this application include the following:

1. A single electric-driven fuel-booster pump for the four-engine installation in the boat.
2. An engine-driven accessory-drive pad for a starter, a fuel pump and governor assembly, lube-oil pump and tachometer generator.

A thorough treatment of the operating troubles encountered on this engine application has been reported (14). In this earlier analysis, which was made at the end of approximately 20,000 hours of operation, it was significant that the forced outage time on the engines amounted to only 4.58 per cent. This gives an engine reliability of 95.4 per cent excluding the downtime for scheduled maintenance. If the time for scheduled maintenance is included then the engine availability is 76 per cent. Considering the fact that this engine was rushed into operation, before detailed and thorough testing had been completed at the manufacturer's plant, this figure of 76 per cent is considered excellent. Industrial applications of large engines have shown an over-all availability of 86 per cent (15).

At the end of December 1954, which is the end of the reporting period just mentioned (14), these engines had a total operating time of 19,933 hours. The engines have now built up an operating time of 31,231 hours. Improvements have been made during this period, including improvements of components and also the turbine. The turbine change has been primarily that of increasing the chord width of the first-stage blades from 0.5 to 0.6 in. This has been done to increase the blade resistance to carbon impact and fatigue. The operation thus far on the heavier blades has been most encouraging (16). The number of blade failures up to 1 January 1955 on the 0.5-in. chord blades was five. Since that time 18 failures have been reported on the 0.5-in-chord wheel. To eliminate these failures all 0.5-in.-chord blades are being replaced with the 0.6-in.-chord and to date none of the heavier chord-bladed wheels has failed in operation. The testing which was conducted in order to assure reliability of the 0.6-in.-chord wheel included a 1000-hour test, the test engine being run 24 hours per day, 7 days per week, at the U.S. Naval Engineering Experiment Station. This engine was run on the following cycles:

15,000 rpm for 1½ hours
25,000 rpm for 1½ hours
36,000 rpm for 6½ hours
Engine shut down for 1½ hours

This cycle was continued for 100 hours and repeated 10 times for a total operation of 1000 hours. Following each 100-hour run, burner liners were inspected for carbon deposit and first-stage turbine-wheel blades were inspected by bore scope. A complete engine disassembly and inspection were made after 300 and 500 hours. At the end of 900 hours, a small crack was noted in one blade; this crack continued to grow, as the test continued and finally caused failure at 1326 hours. Inspection of combustion-chamber liner showed carbon formation at the end of 300 and again at the end of 700 hours. The increase in weight of the burner liners due to carbon formation is shown in Table 4.

TABLE 4

INCREASE IN COMBUSTOR WEIGHT BECAUSE OF CARBON DEPOSIT

Hours run	Combustor weight increase—Carbon deposits (grains) Port	Stbd.
100	1.9	2.9
200	.7	1.4
300	4.3	3.2
400	2.9	1.9
500	1.3	1.6
600	2.6	4.0
700	1.2	3.2
800	2.9	3.1

The 1000-hour engine operation in the Laboratory was completed successfully. The only part replacements that occurred during this time were a starter bearing which was renewed at 300 hours, and an accessory-drive gear; the second-stage power units were installed to reduce the inspection downtime. Following the 1000-hour test the following conditions were noted:

1. The inner shroud ring of the first-stage nozzle box was cracked.
2. The first-stage nozzle box which was of old design was distorted and caused a minor rub.
3. An ignitor plug had failed after about 150 starts were made.
4. Teeth in the second-stage reduction gear were spalled.
5. An oil-seal assembly had a heavy deposit of hard carbon.
6. Bearings in tachometer-drive second-stage reduction showed extensive wear.

This test of the gas turbine would be the equivalent of an automobile traveling at 100 miles per hour for more than 100,000 miles without a major engine overhaul (16).

160-HP MINESWEEPER PROPULSION DRIVE

A new minesweeping launch has been powered by a marine gas-turbine engine. This launch is 36 ft. in length and is powerel by a Boeing 502-8C, 160-hp marine gas-turbine engine. Data on this engine are given in Table 5. Four prototype boats have been completed with this engine installation; three of the boats are equipped with a reverse reduction gear and one boat is equipped with a controllable, reversible-pitch propeller.

The gas-turbine propulsion package consists of the engine and the reverse reduction gear mounted in a rigid rectangular frame. The after portion of the power package has two output sections. The upper

TABLE 5

DATA ON BOEING 502-8C GAS TURBINE

HP	160
Description	Split shaft
Mfgr Designation	502-8C
BUSHIPS Designation	2 MV 27B-8
Weight	319 lbs.
SFC full power	1.37
Length	44"
Width	23.20"
Height	23.34"
Air flow	3.5 lbs./sec.
Fuel	MIL-F-16884A (Diesel oil)
Pressure Ratio	3:1
Turbine inlet temp.	1580°F
Application	Propulsion Generator drive

output section furnishes power to drive the 5-kw ship-service generator, the bilge-pump and salt-water lube-oil-cooling pump. The lower output flange of the reversing gear connects to the propeller shaft. The engine combustion air is supplied through a screened sound-absorbing inlet duct lined with layers of insulating material. Combustion gases from the engine are released to the atmosphere through a duct terminating in an exhaust-muffler assembly which provides for aspirating the ventilation inside the engine enclosure. All controls, instruments, and indicating lights for the engine and reversing gear are located on a remote-control panel.

A description of this boat installation and the boat operating features were given in a paper before the Chesapeake Section, Society of Naval Architects and Marine Engineers, Washington, D. C.—(17). Operations to date has been mainly on the three boats with the reverse reduction-gear installation. The fourth boat with the controllable reversible-pitch propeller has been placed in operation recently. As of 1 November 1955 the operating hours built up on the three boats equipped with the reverse reduction gear totalled 564, and during this time there has been no serious trouble encountered with the turbines. Some minor troubles have been experienced with auxiliaries. Lube-oil sumps which were of aluminum material have been in contact with salt water and serious corrosion problems resulted. These aluminum sumps have been replaced with type 347 stainless steel which has been coated with plastic and it is expected that the corrosion problems have been solved by this change in material. The operating forces have been generally pleased at the way these engines have started and performed during their relatively short service time.

160-HP PROPULSION ENGINE

One of the early marine installations of a gas-turbine engine was made in a LCVP. Following installation the boat was put through an extensive series of tests (19) to determine the suitability of this 160-hp gas turbine for propelling the LCVP. The propulsion plant weight was 820 lb. for the gas turbine and 3040 lb. for the 225-hp diesel. The tests (20) showed the gas turbine to have a number of major advantages, including easier starting, simplicity of maintenance, no warm-up period, and simplification of cooling-water system.

Two Boeing 502-2 gas-turbine engines, Table 6, have been installed and operated in an LCVP at the U. S. Naval Engineering Experiment Station. This installation incorporates dual side exhaust from the engine. The ducts and hot gases from the engine are cooled with salt-water spray before passing through the sides of the boat. Total time on these engines is 180 hours and 836 starts.

The maintenance experience on these engines has been satisfactory. During the last 69 hours of operation, which included 316 engine starts, one igniter plug was replaced; that was the only item on the engine that required maintenance.

160-HP AUXILIARY GENERATOR DRIVE

A gas turbine has been used to power the auxiliary generators in 36 ft. minesweeper launches. Four boats (MSL) have been completed and generator sets driven by a gas-turbine engine, Table 5, have been installed. The MSL has the same type engine for both propulsion and generator drive, hence the number of spare engine parts carried in stock is drastically reduced. This is an outstanding example of a multipurpose application and results in a dollar saving.

The gas-turbine engine and the generator are mounted on a rigid rectangular power-package frame. Three cross members reinforce the frame in the front and the rear near the center. The generator is secured

TABLE 6

DATA ON BOEING 502-2 GAS TURBINE

HP	160
Description	Split shaft
Mfgr Designation	502-2
BUSHIPS Designation	2 MV 27B-2
Weight	236 lbs.
SFC full power	1.35
Length	39.87"
Width	23.20"
Height	22.95"
Air flow	3.5 lbs./sec.
Fuel	MIL-F-16884A (Diesel Oil)
Pressure Ratio	3.1
Turbine inlet temp.	1550
Application	Propulsion

with longitudinal and side members to the frame by four bolts. The gas-turbine engine is in line with the generator and is held in position on the frame by the front engine mount which is bolted to the forward cross brace. An enclosure with a visible control panel is mounted over the entire package and is bolted to the power-package frame. Portions of the enclosure are lined with sound-absorbing fibreglass that absorbs the high-frequency sounds. Exhaust-muffler assemblies have four insulating and sound-absorbing sections which are attached to the enclosure. Air inlet is through a screened section at the forward end of the enclosure and includes a silencer which attaches to the gas-producer case.

Generators have been in use in only three boats up to the present time. A total of approximately 30 hours of operation on the gas turbine have been experienced. No turbine trouble has been reported.

50-HP PUMP DRIVE

The smallest gas-turbine engine sponsored by the Bureau of Ships is the Solar T-45, Table 7. The first application of this engine (22) was to drive a centrifugal pump, with a capacity of 500 gpm 100 psi. Seven of these pump units were procured by the Bureau for test and evaluation, and two units have been given extensive test as portable fire pumps. These gas-turbine-driven pumps have operated as follows: No. 1, 108 hours and 586 starts; No. 2, 101 hours and 220 starts. Maintenance on the units has been negligible. Two pumps are now in the Fleet for operational evaluation.

The use of gas-turbine exhaust gases for deicing (21) exposed equipment and structures on ships in the Arctic is feasible as demonstrated by recent tests. This engine-pump combination was modified to make the engine suitable for deicing. The modification included provisions for attaching a flexible metal hose to the exhaust to pipe the hot gases to areas where icing had occurred. The exhaust hose in this application is 8 in. (diameter) of stainless steel in three sections each 10 ft. in length. Heat release is approximately 1,500,000 Btu per hour. Two units were sent to the Arctic during the 1954-1955 expedition. Evaluation of the unit as a deicer included the following phases:

1. Determine time required to set up the unit and start operation.
2. At the start of severe icing conditions, test the ability to keep winches, reels, hawse pipes, rigging, antennas, and structures free of ice and dry.
3. After icing from 4 to 6 in. depth, test ability to remove ice and dry the surfaces.
4. Determine usefulness in warming up small-boat diesel engines, to aid in starting.

Following evaluation, it was concluded that this gas-turbine engine is outstanding as a portable means for deicing. No difficulties were experienced during starting or operation of the engine. Gas-turbine exhaust for deicing appears to be the best method yet devised when used in conjunction with shovels and the pressure from a fire hose for the final break up and washing away of the slush.

The modification of the engine as a fire-pump drive is simple and inexpensive. Here is a place where "value engineering" comes to the front. With a gas-turbine-driven fire pump on board, the same unit can also be used for deicing. Additional tests are to be conducted on the deicing version with some minor modifications as suggested by the operating forces. It is interesting to note that the Air Force is now using a modified version of the T-45 gas turbine as a generator drive in aircraft. Over 200 units are now used in aircraft and have accumulated over 200,000 hours of operation. Time between overhauls on this engine has recently been increased from 500 to 750 hours.

TABLE 7

DATA ON SOLAR T-45 GAS TURBINE

HP	46
Description	Single shaft
Mfgr Designation	T-45 M-2
BUSHIPS Designation	1 MC4S-1
Weight	65 lbs.
SFC full power	2.2
Length	24″
Width	26″
Height	26″
Air flow	2.4 lbs./sec.
Fuel	MIL-F-16884A
Pressure Ratio	2.45:1
Turbine inlet temp.	1180°F.
Application	Pump Drive Air Supply Deicer Generator Drive Smoke generator

160-HP DEICER

The Boeing 502-6 engine which is rated at 160 hp has been evaluated in Arctic operations using the hot exhaust gases for deicing. The engine characteristics are listed in Table 3. This engine is normally a split-shaft unit and is used in other applications as a generator drive. The deicing version of this engine consists of the following:

1. The first stage gas producer section.
2. A transition cone containing four 2½-in.-diameter nozzles replaces the second-stage nozzle box.
3. Four 4-in.-diameter tubes operating in conjunction with the nozzles which provide an inductor system for entraining ambient air to reduce the high exhaust temperatures to about 750° F.
4. Four 4-in.-diameter 30-ft. lengths of stainless steel flexible hose with quick connectors which are attached to the open end of the inductor section.

The gas-turbine is set inside an enclosure with carrying handles so that it may be transported by two men. Two 10 gallon fuel tanks are supplied but are

not attached to the unit. They are transported to the area of operation and with the quick-disconnect fuel-line attachments can be easily assembled with the unit. The lubricating-oil system consists of a filter, a tank and an air-cooled heat exchanger. The necessary instruments and controls are located in a panel mounted in the top of the unit.

This deicer was tested in the Arctic to demonstrate its ability to deice main-deck superstructures and exposed equipment and also to warm up small-boat engines to make easier starting during cold-weather operations. The scope of the tests which were conducted included the following:

1. Determine the time required to break out the gas-turbine deicing unit, locate it on deck, start the engine and place it in operation. During this phase, the unit was cold-soaked on deck over night and then started early in the morning.
2. Under the start of severe icing conditions, test the ability of exhaust from this unit to keep top-side equipment such as capstans, winches, reels, hawse pipes, rigging, antenna and horizontal and vertical surfaces free from ice and dry.
3. After icing had taken place from 4 to 6 in. in depth, test the ability to remove this ice.
4. Conduct a test to determine this unit's ability to warm up small-boat diesel engines.
5. Compare the present method of deicing which requires the use of sledges, bars and hammers with the gas-turbine-exhaust deicing procedure.

Testing of this unit was conducted on the CGC *North Wind* during February-April 1955. The unit was set up on the forcastle by four men in about 20 minutes. It started immediately with the outside temperature at 19°F. Some trouble was experienced due to freezing up of two of the exhaust lines which had been lying idle. These two lines were disconnected and the turbine was then operated for about 10 minutes until the exhaust lines were melted free of ice. The next day the unit was again started after having been left in the open over night, during which time the temperature had fallen to 9°F. The temperature at the time of starting was 15°F. This test included operation of the unit with all four exhaust hoses. A considerable amount of equipment and the deck structure forward of the 5-in. gun mount were completely freed of ice, or ice loosened so that it was later easily removed by a fire hose. This engine was not used for heating small-boat engines for cold-weather operation because of the high exhaust temperature which was approximately 750°F. It is concluded that deicing by means of a gas-turbine exhaust is far superior to deicing methods used in the past. When the deicing is done by gas-turbine exhaust and in conjunction with personnel using shovels, it is a very attractive method of freeing topside structures and equipment from ice. Where hammers and bars in the vicinity of electrical and hydraulic equipment cannot be used, the gas turbine is even more attractive.

NEW CONSTRUCTION APPLICATIONS OF GAS TURBINES

It is now evident that the gas turbine has demonstrated its reliability in the marine and naval fields. Experience has been obtained by the Navy on the marine gas turbines and a number of new construction projects, Table 8, are under way that will use gas turbines as propulsion and auxiliary drive. The following programs are now active:

1. Twenty-six minesweeping launches are being procured. These launches will be propelled by gas-turbine engines and will also have auxiliary generators which are driven by gas turbines.
2. Ten new landing craft are being procured. They will have gas-turbine engines for propulsion units.
3. A destroyer conversion is currently under way which will use a gas-turbine-generator set.
4. A 40-ft. personnel boat is being propelled with a 500-hp gas turbine.
5. A PT boat is having two 4000-hp gas-turbine engines installed for propulsion. One gas-turbine-driven fire pump will be placed on board.
6. A beach lighter is being equipped with a gas-turbine-driven fire pump.

TABLE 8

SUMMARY OF ENGINES BEING PROCURED FOR INSTALLATION IN NAVAL VESSELS

Rating HP	Application	Vessel Type	Vessel Number	Number of Engines	Total Horsepower
46	Fire Pump	Beach Lighter	1	1	46
46	Fire Pump	PT Boat	1	1	46
46	Auxiliary	Minesweep	48	48	2,160
220	Propulsion	Minesweep	26	26	2,720
220	Auxiliary Generator	Minesweep	28	28	5,720
4000	Propulsion	PT Boat	1	2	8,000
500	Propulsion	Personnel Boat	1	1	500
500	Emergency Generator	Destroyer	1	1	500
		Total	107	108	22,748

BIBLIOGRAPHY

1. "British Naval Gas Turbines," by G. F. A. Trewby, presented before Joint Meeting New York Metropolitan American Society of Mechanical Engineers and New York Metropolitan Section of Naval Architects and Marine Engineers, April 27, 1954, and before Washington, D.C. Section of The American Society of Mechanical Engineers, April 28, 1954.
2. "A 5000 KW Railway-Mounted Gas Turbine Power Plant," by C. H. Giroux, J. O. Stephens, and R. J. Nolte, ASME Paper No. 54-A-191.
3. "Brevities of Gas Turbine Interest," The Oil Engine and Gas Power, vol. 23, no. 269, November, 1955, p. 282.
4. "Belle Isle Gas Turbine: After 30,000 Hours," by J. W. Blake, Power, vol. 98, August, 1954.
5. *Wall Street Journal*, December 1, 1955.
6. "Direct-Drive Gas-Turbine Unit for Tanker," Engineering (British), September 9, 1955, p. 324.
7. "The Gas-Turbine Installation in Liberty Ship John Sergeant," by J. J. McMullen, Trans. SNAME, vol. 63, 1955.
8. "An Automatically Controlled Ship's Stand-By Generator Set," The Oil Engine and Gas Turbine, vol. 23, no. 266, mid-August, 1955, pp. 156-158.
9. "The Effects of Offshore Procurement," *Aero Digest*, vol. 71, no. 4, October, 1955, pp. 32-34, 36.
10. "Shell and the Gas Turbine," Gas and Oil Power, March, 1954.
11. "Design Features of a 250 KW Gas Turbine Engine," by R. R. Peterson and P. G. Carlson, Mechanical Engineering, vol. 74, March, 1952.
12. Designation System for Gas Turbine Engines, Marine, MIL-STD-213 (Navy) July 28, 1953.
13. "The Development of a 500-HP Multipurpose Gas Turbine Engine," by P. G. Carlson and I. M. Swatman, ASME Paper No. 55-S-36.
14. "Evaluation of a Naval Gas Turbine Engine," by J. A. Culver and A. L. Wood, paper presented at the Diamond Jubilee Spring Meeting, Baltimore, Md., April 18-22, 1955, of The American Society of Mechanical Engineers.
15. "Gas Turbine on The Line 27,000 Hours (A World's Record)," by J. W. Blake, Electric Light and Power, May, 1953.
16. "1000-Hour Gas Turbine Test," by W. M. M. Fowden, Jr., Bureau of Ships Journal, vol. 4, no. 6, October, 1955, pp. 19-22.
17. "U. S. Navy's Gas Turbine Propelled Minesweeping Launches," by W. M. M. Fowden, Jr. and R. W. McAleer, paper presented before Chesapeake Section of The Society of Naval Architects and Marine Engineers, Washington, D.C., January 25, 1956.
18. "Test of a Boeing Gas Turbine for Propulsion Service in an LCVP," U. S. Naval Engineering Experiment Station Report 31 (1) 966706, March 3, 1953.
19. "The Gas Turbine as a Prime Mover on U. S. Naval Ships," by W. M. M. Fowden, Jr., R. R. Peterson, and J. W. Sawyer, ASME Paper No. 53-A-233.
20. "Installation and Operating Experience With a Naval Gas Turbine Powered Landing Craft," by J. O. White and J. S. Passman, paper presented at the Semi-Annual Meeting, Pittsburgh, Pa., June 20-24, 1954, of the American Society of Mechanical Engineers.
21. "Deicing by Gas Turbine Pump Rated," by D. E. Blackwood, Bureau of Ships Journal, vol. 3, March, 1955, pp. 7-9.
22. "The T-45 Power Plant," by R. R. Peterson and A. D. Zakarian, *Journal of the American Society of Naval Engineers*, vol. 63, August, 1951, pp. 555-567.

ARNOLD BUCHSBAUM

REVIEWING DRAWINGS FOR BETTER SHIPS

THE AUTHOR

is a supervisory marine engineer at the New York Naval Shipyard. The article is based on his experience of more than eighteen years in various grades at engineering from the drawing board and up. His specialty is in internal combustion engine work. Mr. Buchsbaum is a graduate in mechanical engineering from the College of the City of New York and has taken graduate courses at that College and at Columbia University. He is a member of the American Society of Naval Engineers.

A GOOD DRAWING is the foundation of a well built ship. To produce a perfect set of drawings for that vast complex of structures, equipment, and systems making up the modern naval vessel is a feat of legerdemain normally associated with infallible beings. Unfortunately, engineers, designers, and draftsmen, being basically human, are rarely infallible. In their efforts to create, they will err, promote pet schemes, indulge in daydreams, and occasionally suffer from nightmares. When such aberrations from infallibility are impressed on a drawing and manufacturing ensues, the results could be ludicrous, regrettable, or catastrophic, depending on the nature and extent of the fallacies committed.

In order to eliminate these fallacies or at least mitigate their evil effects, it is customary, if not essential, to review each drawing before any manufacturing is undertaken. Unhappily, the review costs money; time and effort are expended; those relentless drawing and production schedules must be met or appeased.

How much time should be taken? How broadly, how deeply, and how intensely should each aspect and detail of the drawing and design be reviewed? How can minimum costs be achieved without sacrificing accuracy and integrity? Who should do the reviewing, what qualifications are necessary and how should the reviewer be fitted into the organization of the design office for maximum efficiency?

Naturally, the less reviewing or checking that is done, the less will be the overall cost of the drawing and the easier it will be to meet committments imposed by the normally tight planning schedules. On the other hand, if a practice of minimum review is indulged in excessively, it may lead to repeated and serious errors, which could actually increase the overall cost of production, and may generate serious customer dissatisfaction and disdain. At the worst, it might result in a permanent loss of clientele or of the business itself.

How can one arrive at a judicious balance between maximum and minimum review? Unfortunately as in most technical arts where intuition and experience are overriding guides, it is impossible to set up exact rules for measuring the quantity and quality of review necessary in any particular instance. However, there is little question that the practice of the minimum review as the trend towards simplified draftsmanship is gaining increasing acceptance. The present day common use of the words "review" and "reviews" implying a generalized survey, in preference to the old-fashioned appelations of "check" and "checkers" connoting a microscopic search, merely emphasizes this tendency. However, in order to avoid semantic digressions, the aforementioned words will be used interchangeably here.

Aside from this current design office philosophy favorable towards the lesser review, many other factors enter into the thoroughness with which a drawing is checked, for design and delineation. The past performance, accuracy, and technical "knowhow" of the originator of the drawing may be very influential in determining the effort to be expended in reviewing his work. The use of mock-ups, visual aids, models, the availability of information on similar designs, the necessity for meeting drawing issue dates, the extent to which the production shops are accustomed or allowed to propose, create, or guess for themselves, the number of standards and standard drawings which can be made to apply, all bear importantly on the time required for reviewing the drawing.

The availability of the actual ship or equipment for such purposes as installation checking rather than the more complicated visualization and reconstruction necessitated by the use of numerous reference drawings will definitely affect the checking time expended. The extent to which reference drawings themselves can be trusted for accuracy, currency, and completeness will certainly be a factor to consider. Designs linked to reference drawings of uncertain antecedents and apocryphal authority may (and have) come to grief because of lack of appreciation

of this factor. It needs only the headaches involved in applying designs to some assumed "sister ships" to emphasize this point.

In determining the optimum checking effort, one must also draw aside the blue print curtain of the design office and peer beyond, into the manufacturing world itself, since production requirements may importantly affect the economic aspects of the checking work. Thus, a mass-produced item, or a very unique item where high manufacturing and materials costs are involved would warrant the expenditure of a good deal more checking time than an item where a few minutes labor with the welding torch and grinding wheel would put everything in the best order again. "Make to suit," or "make to fit" may be fine in some cases but may be unsuitable or unfitting in other cases. On the other hand, many laborious determinations and checking of difficult drafting problems such as intricate intersection of complicated castings may have no beneficial effect in reducing the shop work but may be costly to the design office. Solving these problems could add to one's luster and self-respect as a perfectionist but may be useless and expensive if the shop man in the natural course of his work can either more readily determine the answers to these problems himself or may find the information quite superfluous.

In other words there must be a constant balancing between the requirements of the design office and the shops. Excess checking is useless, time consuming, and schedule retarding. A superficial or slurred review, while saving checking time, may result in misleading or contradictory information which would require shop stand-by time until the situation is untangled. This would lead to an overall loss of economy and productivity not compensated for by the gains in the design office. In many respects, checking or reviewing may be likened to insurance in that most everyone agrees that it is a fine thing to have but very few feel certain as to what the exact amount should be.

Since the review is a process of judicious, painstaking and detailed determinations, it can be seen that the reviewer must be a person with some superior qualifications. He must also be properly welded into the organization of the design office as a co-operative adjunct. How he is chosen, and how he fits into the organization depends to a large extent on the size, type, and structure of the design office.

In the smallest office, it is possible for the checker, designer, engineer, and draftsman to be one person. In the largest office with an extensive system of checks and reviews along the line, it is not improbable for the checker to be limited in his functions to that of drafting proofreader. It is considered, however, that for the general case, the checker is one who has the capacity and does check the design, engineering, and drafting aspects of a drawing. In this respect, he should, therefore, not only be able to ferret out errors, but also to suggest improvements in the design or product.

The professional and personal qualities which the successful checker must possess depend to a large extent on the type of work he performs. Therefore, in addition to being technically trained and experienced, he should be a model of accuracy, judicious detachment, alertness, patience, conscientiousness, and single-mindedness. He should, also, have the ability to discriminate between important and minor details and the amount of effort to be expended on each. Lastly, and very importantly, he should be susceptible to new ideas, and able to get along harmoniously rather than acrimoniously with others in the office, especially the draftsmen.

In fitting the checker into the organization of the office, two general schemes are possible. In one, the checker is assigned possibly as a squad boss to a group of men and checks their output exclusively. In the other scheme, the checker is part of a checking section, exclusive of, and generally compartmentalized from the rest of the office.

The first scheme allows for a close check of the work at the source, makes the possibilities of major revisions or additions after checking less likely, and allows for a good deal of cooperation, explanation, or give-and-take between checker and draftsman. On the other hand, it throws duties other than checking on the checker, does not permit him the full powers of concentration on one job, and may bring in personal predilections insofar as his contacts with the draftsmen are concerned.

The second scheme, although losing by the lack of direct contact, gains in that the checker is freed of supervisory duties, can fully concentrate on his job, and working together with other checkers, can improve his knowledge of and his methods of checking. In addition, the supervision of the checking work is simplified, and the possibilities of work standardization, with attendant benefits, are increased.

Whether either scheme, however, or a combination of both is used, the lines and limits of authority should be clearly defined and all drafting work standardized to the greatest possible degree. Otherwise, endless disagreements might ensue between the checker and the draftsman as to the extent of the revisions or additions permissible to the checker, where clear cases of error cannot be established. After all, it must be remembered that, despite some contrary opinion, the checker is human, too, and possesses his small share of the pet whims, fancies, schemes, and ideas common to lesser humanity.

In the actual performance of his work, the checker generally has the use of a blueprint of the unchecked drawing to indicate thereon the revisions, additions, deletions, or other drawing changes required. Notations are shown in colored pencils or crayon, and the colors chosen to contrast most strongly with the blue background and white lines of the blueprint. In this respect yellow and red are the preferred colors, some checkers using yellow to indicate correct items and red to indicate changes, while others prefer it the other way around. Another common method is to use a yellow crayon to obliterate an unwanted item and

then write over the yellow patch with an ordinary black pencil. Whichever methods are used, they should convey to the draftsman clear and concise information as to changes required. If the checker expects precise results from the draftsman it is only fair that the draftsman receive precise information in turn.

Since there are no generally accepted standards in respect to the use of preferred colors and type of check prints, it might be advantageous for the design office to effect some standardization in this field so that each draftsman will be exactly conversant with each checker's notations and meanings. In offices where a great deal of written matter customarily appears on the drawing, the adoption of standard proofreading symbols to the checking work might advantageously be studied.

The actual approach of the checker to his work and the investigating methods he may use in the course of it will depend largely on his personal work habits, his professional abilities, the extent to which he mirrors the ideals of a perfect checker, and the type of work under consideration. Therefore, it cannot be said that there is an iron-clad or even best way of checking a drawing; each job may be unique and require individual treatment. Without question, however, the work should proceed in orderly and rythmic sequences, such as from a general survey, to the installation, to the assembly, to the detail, to the bill of material, to the notes, back-checking to the design requirement, and so forth. The idea of checking everything in all directions at once should be definitely discouraged.

To aid the checker in attaining the optimum conditions of accuracy and speed, the design office should contain a good library, well stocked with a carefully chosen set of reference books, pamphlets, and magazines, and cross-indexed for ready reference. It should also contain as complete a set of design and drafting standards as are possible for the type of work handled. In addition, its plan files should be so ordered that the invaluable experience to be gained from plans already issued and successfully used should be readily available for reference, example, and study.

The use of a check list such as the one appended should prove of great benefit to the checker as a memory refresher and guide for the orderly pursuit of his work. Although such a list cannot provide for the entire field of draftsmanship and design, it can obviously be made up to reflect the high points of the checking work required. Thus, separate lists or appendages to a basic list can be established for electrical, piping, structural, electronic, and other specialty work. (The list given herein is a basic list, mildly biased towards naval mechanical needs.)

Finally, the habitat of the checker should be carefully studied for adequate lighting and space to insure a humming contentment on his part. It is certain that he needs well lighted surroundings in the same way as a draftsman does. In so far as possible his space allotment should be as generous as can be obtained within the drafting office confines. A reference table is a valuable aid to his work. It is a pitiful sight to see a checker buried somewhere between the fifth and the fiftieth reference drawing, struggling with one hand to keep from being crushed beneath the references, using the other hand for checking the drawing, with books, pencils, erasers, pads, and rulers scattered about—all on top of a three-foot by six-foot work table. If space is so tight that individual reference tables are not feasible, then a reference table assigned to a small group of checkers can be used to advantage as the most realistic compromise.

When the checking has been completed and the corrections made, the original drawing and the checkprint are returned to the checker for review and signature. At this point great pains must be taken to insure that the required corrections have been properly made and that new errors have not been introduced thereby. It is not impossible for the draftsman in the course of making the changes to inadvertently erase a line, blurr a dimension, alter a detail, or commit other errors in those parts of the drawing that require corrections adjacent to or superimposed on correct matter. These mistakes are insidious and often difficult to find. The best prescription against them is a very careful recheck, luck, prayers, and the realization that checkers, like other humans, are not infallible.

A CHECK LIST FOR CHECKERS

NOTE: This list cannot by any means be considered all inclusive. It does include, however, some standard drafting practices which "Functional Drafting" would castigate, if not exterminate.

1. *Check General Design*
   Have the following design features been properly considered?
   - (a) Adequate strength (including shock and fatigue factors)
   - (b) Sufficient rigidity
   - (c) Correct shape or contour
   - (d) Suitable material (Strength and rigidity; corrosion, wear, and abrasion resistance; stability, machineability, and availability)
   - (e) Weight requirements
   - (f) Practical manufacture
   - (g) Standardization
   - (h) Maximum manufacturing economy
   - (i) Pleasing appearance
   - (j) Suitable operating provisions such as
     1. Interference elimination (Suitable running clearances)
     2. Lubrication
     3. Drainage
     4. Vibration control
     5. Safety devices (eliminate personnel hazards and machine breakdowns)
     6. Comfortable operating conditions (minimize personnel discomfort—habitability)
     7. Allowance for thermal expansion and contraction
   - (k) Ease of assembly and installation
   - (l) Accessibility for in-service inspection and repair
   - (m) Adequate provisions for wear allowance

(n) Provisions for tooling and inspection
(o) Conformity to contract specifications, detail and special specifications, shipalts, circular of requirements, General Specifications for ships, General Specs for inspection of materials, Bu-Ships Manual, Federal and military specifications and standards

2. *Check Drawing in General*

Does a general survey of the drawing indicate the following?
Does it adhere to the general drawing practices of JAN STD 1?
(a) Good layout
(b) Neatness
(c) Legibility
(d) Sufficient information
(e) Simplicity and clarity

3. *Check Drawing in Detail*

Does a detail study of the drawing show the following?
(a) Correct size of sheet (MIL STD 2A)
(b) Correct layout of sheet (proper placing and correct sizing of title block, list of material, notes, reference plans, alterations block, as per MIL STDS 3A & 4A)
(c) Sufficient number and clearness of views and sections to adequately describe the object, without unnecessary duplication and excess line work
(d) Adequate identification of each view and section (Left and right hand items clearly distinguishable)
(e) Correct angle of projection
(f) Correct line work, such as
1. Sufficiently dark lines (for readable blueprints)
2. Proper line contrast (right width and character)
3. Center lines where required
4. Minimum number of hidden lines
5. Conventional drafting symbols (cross-hatching, screw-threads, mechanical, welding, non-destructive tests, according to JAN STD 1, MIL STD 9, 17, 19, 23A respectively)
6. Adequately large letters and figures
(g) Correct dimensioning (MIL STD 8A)
1. Sufficient dimensioning for manufacturing and inspection purposes
2. No duplicating or ambiguous dimensions
3. No additions, subtractions, or other computations left to shops
4. All dimensions in accordance with design requirements
5. All overall dimensions given
6. Appropriate identification when needed for clarity of such dimensions as diameter, radius, diameter of pitch circle, square, hexagon, distance across flats, angle, spherical diameter, and spherical radius
7. Centers of all radii indicated
8. No finished surfaces or centers of finished holes dimensioned from rough surfaces
9. All dimensions correctly scaled and off-scale dimensions indicated
10. Tolerances and allowances in accordance with best usage (including tolerances on fractional dimensions, out-of-roundness, run-out, parallelism or squareness of surfaces and axes)
11. Correct screw thread identifications and fits (Federal Handbook H-28 & Supplement)
12. Clear identification of gauges such as sheet metal, wire, drill
13. Dimensions located outside of detail where possible
14. All dimension lines ending in arrows at related extension lines
15. All extension lines correctly aligned
16. No extension lines touching aligned surfaces
(h) Correct use of finishes and finish marks (MIL STD 10)
1. All finished surfaces and type of finish indicated by suitable symbols
2. Allowances for finish given where required
3. Finishes to be a minimum consistent with design requirements
(i) Adequate list of material
1. Correct part number
2. Descriptive title for part
3. Amount required given
4. Material in accordance with design requirements
5. Standard material specification and stock numbers used wherever possible
6. Weights given where required
(j) Complete set of notes, including the following:
1. Standard tolerances when given by notes
2. Test pressures
3. Heat treatments including stress relieving
4. Magnetic and fluorescent particle inspection, radiography, reflectoscope
5. Material or parts substitutions or alternates
6. Cleaning, painting, plating, shot peening, preserving, etc.
7. Breaking sharp corners and edges
8. Spotfacing for bolt heads and nuts
9. General explanatory notes as to manufacture, assembly, and installation
10. Superseding drawing notes
11. Design or operating data given when required by contract
12. Operating and safety precaution instructions
(k) Adequate references
1. All pertinent reference plans listed
2. Forging die numbers, heat treat numbers, pattern numbers, jigs, fixture and tool numbers listed where available and required
3. Items ordered from catalogues fully identified
(l) Correct and complete title block including:
1. Fully descriptive title
2. Scale of drawing
3. Angle of projection if other than standard
4. Date of drawing
5. Signatures
6. Plan number and revision number
(m) Proper abbreviations (MIL STD 12A)

4. *Recheck Drawing After Corrections Have Been Made*

(a) Have all the corrections been properly made and have no new errors been introduced?
(b) Are Standard Revision symbols being used and do they clearly identify the revisions? (MIL STD 24)

H. S. PREISER and M. STANDER

APPLICATIONS OF PLASTICS IN MARINE CATHODIC PROTECTION SYSTEMS

HERMAN S. PREISER

Corrosion Engineer, Coatings and Preservation Branch, Bureau of Ships. A graduate of the New York State Maritime Academy in 1946 as a licensed engineer, he received a B.S. Degree in Naval architecture and marine engineering from Webb Institute in 1949. He has had graduate work in electrochemistry and metallurgy at George Washington University and University of Maryland. He is a member of the Society of Naval Architects and Marine Engineers, The Electrochemical Society and the National Association of Corrosion Engineers. He holds a license as Registered Professional Engineer in the District of Columbia. His early work at the Bureau of Ships in 1950 led him to Portsmouth Naval Shipyard where he took an active interest in corrosion protection of submarines. Upon returning to the Bureau of Ships, he specialized in submarine design. In 1952, Mr. Preiser transferred to Coatings and Preservation Branch, Research and Development Division, as a corrosion engineer. At present, he is in charge of research and development work on cathodic protection for the Branch. His work has brought him in intimate contact with practical corrosion problems of the fleet.

MAXWELL STANDER

is a Materials Engineer (Plastics) at the Bureau of Ships. Was formerly Assistant Research Coordinator for Materials in the same organization.

He has a B.S. in chemistry from the College of the City of New York and has studied Chemical Engineering at New York University and George Washington University. He has been employed as a Fuels and Lubricants Chemist with the Paragon Oil Company and Chemical Engineer at Advance Solvents & Chemical Corporation. Prior to his present position, he was employed as a coatings and materials chemist with the National Bureau of Standards and other government organizations. He is a member of the Society of Plastics Engineers and the Armed Forces Plastic Boat Committee.

NOTE: *This paper was presented at the Society of Plastics Engineers National Technical Conference at Cleveland, Ohio, on 19 January 1956.*

INTRODUCTION

CATHODIC protection techniques for corrosion control of marine structures are gradually finding widespread acceptance in marine and naval fields. Briefly stated, cathodic protection is the use of an impressed direct current for reducing or suppressing electrochemical corrosion of a metallic structure immersed in an electrolyte by making the structure a cathode for the impressed current. The impressed current may be supplied by sacrificial galvanic anodes or by rectifier energized anodes. This paper will confine itself to the latter system employing inert anode materials.

CORROSION OF SHIP HULLS

Before we get involved in the finer points of component design it might be well to examine the factors responsible for the corrosion of the underwater hull of a ship and learn just how cathodic protection can help us out of our difficulties.

Bare hull steel immersed in seawater corrodes at the average of 5 mils per year. This fact applies to most carbon steels anywhere in the world.[1] If the corrosion of the steel were distributed uniformly over the metal surface, the problem would be of little consequence. Unfortunately, this is not the case. Most steels have a natural tendency to pit. This re-

sults in concentrated localized attack of five to ten times the severity of the average corrosion rate and is a problem of major concern.

Since ship hulls are generally painted, imperfections in the coating such as holidays, porosity and the physical breakdown of paint itself expose bare areas which are susceptible to pitting. Pitting is basically an electrochemical effect where the solution of small localized anodic areas is encouraged when certain conditions exist in seawater or at the exposed surface of the metal. For example, the pitting observed on mild steel is partially the result of heterogeneity of the steel. Relatively minor differences in chemistry, cold work, heat treatment, non-metallic inclusions, and so on produce many localized and susceptible anodic areas. Velocity effects, varying oxygen content of seawater, galvanic couples of dissimilar metals and the presence of mill scale contribute further to the pitting of the metal (hull). These types of electrochemical attack can be countered by the combined use of cathodic protection and effective paints.

CATHODIC PROTECTION

Cathodic protection may be visualized as a reversed galvanic cell, where sufficient current supplied from external source (auxiliary anode) is impressed on a metallic structure (cathode) to raise the potential of the system cathodes to that of the system anodes (polarization). The nullification of the potential differences between the anodes and cathodes in the system will prevent the flow of self-generating corrosion producing currents within the structure.

In other words, if more current is caused to flow into a corroding area than has a tendency to leave, for all practical purposes electrochemical corrosion will cease.

Fortunately, there is a simple method of determining whether or not a metal structure is corroding or is being protected. Every metal has a freely corroding galvanic voltage which is called its half-cell potential. A list of half-cell potentials in seawater has been compiled by arranging metals in their decreasing order of activity. Table I is a galvanic series of commonly used metals and lists magnesium (anodic) at the top and nickel (cathodic) at the bottom. It is general knowledge that a galvanic cell is the electrical combination of two half-cells immersed in a common electrolyte. The galvanic voltage associated with such a cell is the algebraic difference of two half-cell potentials. The total voltage of the cell can be measured directly by connecting the half-cells in question through an appropriate potentiometer.

By knowing the value of any one half-cell, the other can be calculated readily. For cathodic protection work, there are available standard reference electrodes which are merely specially constructed half-cells of known stable values. Two common types are in use: the copper-copper sulphate and silver-silver chloride reference electrodes.

TABLE I

Galvanic Series of Common Metals in Seawater

Anodic	Magnesium
↓	Zinc
	Aluminum
	Cadmium
	Lead
	Tin
	Nickel (Active)
	Copper
Cathodic	Nickel (Passive)

A particular metal in the series will be anodic to all metals listed below it and cathodic to all metals listed above it.

The flow of current to a metal causes its half-cell potential to change. A protective potential is reached when the current flowing polarizes the local cells to equilibrium. This is the established criterion for total cathodic protection; namely, the polarization of local cathodes to the open circuit potential of the local anodes. For steel which normally corrodes at —600 mv, polarizing it to —800 mv (referred to Ag-Agcl half-cell) will provide almost 100% electrochemical corrosion protection.

CHEMICAL REACTIONS ASSOCIATED WITH CATHODIC PROTECTION

The chemical reactions associated with steel under cathodic protection in seawater produces a calcareous deposit on the metal surface which is alkaline in nature. Seawater contains large quantities of Na^+, H^+, Cl^-, OH^-, Ca^{++}, and Mg^{++} ions in addition to a host of others. Since the cathode is negatively charged in a cathodic protection system several possible reactions occur:

(1) $H^+ + e^- \rightarrow H^0$
(2) $2H^0 \rightarrow H_2$ or
(3) $2H^0 + \frac{1}{2} O_2 \rightarrow H_2O$

Reaction (1) is the plating out of atomic hydrogen on the cathode. Reaction (2) is the conversion of atomic hydrogen to hydrogen gas (overvoltage) which then bubbles from the cathode allowing more hydrogen atoms to plate out. Reaction (3) is the removal of hydrogen by oxygen depolarization; the more common reaction in aerated seawater. The rapid removal of H^+ ions from the vicinity of the cathode leaves a concentration of OH^- ions. This can be summerized by:

(4) $2H_2O + 2e^- \rightarrow H_2 + 2\ OH^-$

The hydroxyl ion-rich electrolyte then readily combines with Mg^{++} and Ca^{++} ions to form a precipitate which is thrown down on the cathode.[2]

(5) $Mg^{++} + 2(OH)^- \rightarrow Mg(OH)_2$
(6) $Ca^{++} + 2(OH)^- \rightarrow Ca(OH)_2$

Another possible reaction is:

(7) $Mg^{++} + 2e^- \rightarrow Mg^0$
(8) $Mg^0 + 2\ H_2O \rightarrow Mg(OH)_2 + H_2$

and similarly for calcium

(9) $Ca^{++} + 2e^- \rightarrow Ca^0$
$Ca^0 + 2\ H_2O \rightarrow Ca(OH)_2 + H_2$

These precipitates are further hardened into the familiar calcereous deposits by the dissolved CO_2 in the seawater.[3, 4]

(10) $4\ Mg(OH)_2 + 3CO_2 + H_2O \rightarrow Mg(OH)_2 \cdot 3MgCO_3 \cdot 4H_2O$
(11) $Ca(OH)_2 + CO_2 \rightarrow CaCO_3 + H_2O$

The anode reactions associated with insoluble anodes such as graphite or platinum in seawater are as follows:[5, 6]

(12) $2Cl^- \rightarrow Cl_2 + 2e^-$ or
(13) $Cl^- + 2\ OH^- \rightarrow ClO^- + H_2O + 2e^-$
(14) $Cl_2 + H_2O \rightleftarrows (H^+ + Cl^-)\ HOCl$

WHY PLASTICS ARE NEEDED

It should be apparent from the foregoing discussion that specific materials are needed for use in cathodic protection systems. The particular properties sought would depend on the end use of the material. In general, materials are needed which are non-conducting, alkali resistant, oxy-chloride resistant, have low permeability to water transmission under a potential gradient, low water absorptivity, are non-reactive to seawater, have high strength, are impact resistant and are readily formed into a variety of shapes. Plastic materials because of their versatility lend themselves admirably for components of cathodic protection systems.

COMPONENTS OF CATHODIC PROTECTION SYSTEMS

Briefly, the elements of a cathodic protection system consist of the following:

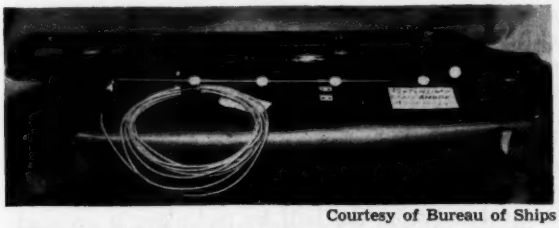
a) The anode
b) The cathode
c) Current source and External circuit
d) Measuring instruments

The anode for impressed current systems is usually a platinum-clad silver or copper rod or a graphite rod or slab. The anode must be mounted in suitable holders which must be dielectric, oxy-chloride resistant, and rugged.

The cathode of a cathodic protection system is the structure under consideration; in our case a ship's hull. In order to insure adequate current distribution from the anode, shield materials are mounted on the hull in the vicinity of the anode. These materials must adhere to steel, have low permeability to electro-osmosis, have alkali resistance, be tough and flexible, and have good dielectric properties.

The current source and external circuit implies the use of chemically resistant cable sheaths and suitable cable attachment to the anode with appropriate seals.

The measuring instrument used for determining the cathodic potential of the structure is the reference electrode. This sensitive element must be housed in a dielectric, non-reactive, streamlined casing which is fitted with suitable cable seals.



Courtesy of Bureau of Ships

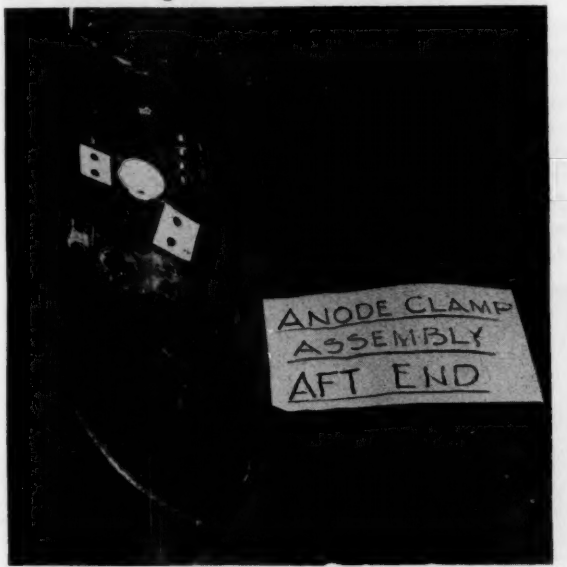
Figure 1. Porcelain enameled steel holder assembly for a platinum clad anode (unassembled).

The remainder of the paper will deal with specific designs of the various components and the future role of plastics in the field of cathodic protection.

PREVIOUS PLATINUM-CLAD ANODE ASSEMBLY

Figure 1 shows the platinum-clad anode assembly in current use in the Navy. The assembly consists basically of a metallic rod capable of carrying appreciable currents mounted in an insulated inverted tee section support. An appropriate cable seal in a stuffing tube assembly connects the anode rod to the current source. The rod is made of coin silver over which is clad a 5 mil pore free thickness of 90–10 platinum-palladium alloy. The overall dimensions of the rod is ¼″ diameter by approximately 50″ length. The support proper is fabricated from steel over which is applied a two coat system of acid resistant porcelain enamel. In order to provide positive insulation of the rod from the holder, a special clamp arrangement holding "Teflon" discs are provided. The details of the clamp assembly are shown in Figure 2.

This type of holder, in addition to its complex fabrication and high cost has several other inherent



Courtesy of Bureau of Ships

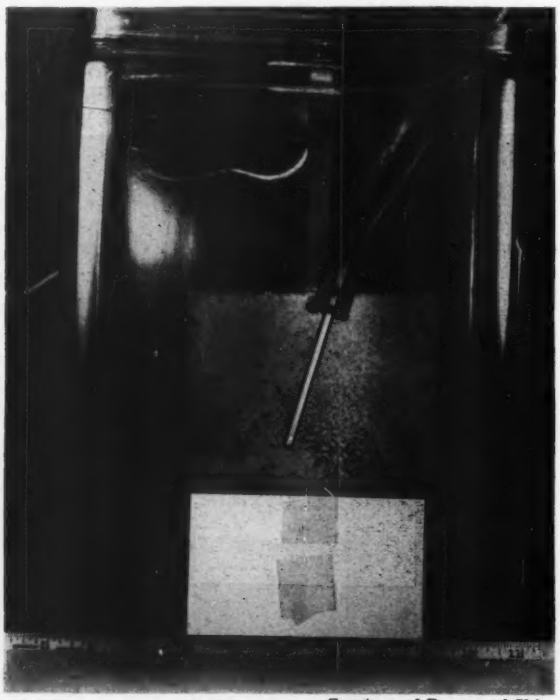
Figure 2. Detailed view of "Teflon" disc and clamp arrangement.

disadvantages. The porcelain enamel coating is difficult to apply free of defects and because of the general brittleness of this type coating, it is susceptible to damage when handled roughly.

Foolproof insulation of the holder is desired for two reasons. One is to promote optimum current distribution from the anode and the other is to prevent e'ectrical damage to the holder. If the holder is insulated from the hull by mounting it on studs which are fitted with insulating sleeves and washers, the holder will not receive cathodic protection. As such, no excess build up of current would occur at bare areas of the holder but a certain amount of stray current would pass through holder and cause pitting of the base metal wherever current was permitted to leave. On the other hand, if the holder was electrically grounded to the hull, no corrosion would occur at bare spots but excess current will flow from anode to holder with the resulting interference with good current distribution. Further, excess calcareous deposits building up on bare spots would adversely affect the acid resistant enamel. Difficulty has also been experienced in mounting this type holder on the curved side of a ship. Any flexure of the holder will cause enamel to chip or crack at areas of compression.

PORTSMOUTH EXPERIMENT OF OXYCHLORIDE RESISTANCE OF PLASTICS

The selection of a plastic or combination of plastics for anode holder components is expected to overcome the difficulties cited above by providing positive dielectric support and flexibility in attachment. Further, because of the lightness, moldability, workability and excellent chemical resistance, a suitable less expensive holder can be fabricated. The newly designed holder would eliminate "Teflon" inserts. Although "Teflon" has excellent oxychloride resistance, its cold flow properties under compression, high cost and inability to be bonded to other plastics make it generally undesirable for inclusion in anode holder construction.



Courtesy of Bureau of Ships

Figure 3. Portsmouth Naval Shipyard apparatus for testing Oxy-Chloride resistant materials.

It was necessary to run some preliminary experiments to determine what plastics are resistant to oxychloride attack.[7]

A series of tests were run at Portsmouth Naval Shipyard to find a suitable plastic material for platinum-clad anode cable seals. A series of ¼" diameter x 4" length platinum-clad anodes were each attached to a length of cable. The electrical junction of each assembly was insulated with a cast molded plastic seal. Several plastic formulations were tried. Each specimen was immersed in a battery jar containing a seawater electrolyte at about 80° F. and connected to a steel plate cathode. A constant D.C. current of 8.5 amperes at 32 volts was impressed on the anode. The seawater was changed every 24 hours to maintain normal salinity and pH. Turbulence was provided by bubbling air through the bath. Figure 3 shows the experimental setup. The seals were examined periodically and the following results noted:

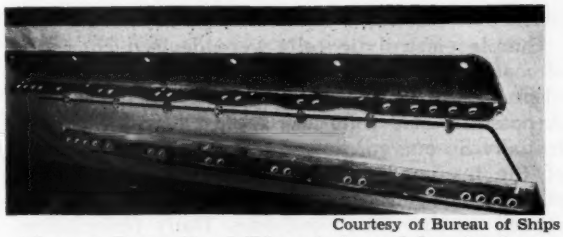
a) Polysulphide rubber (Navy Formula 112)—1/16" thickness dissolved completely in 3 days.
b) Neoprene (cable sheath)—Heavy chalking in 14 days; 1/16" sheath completely destroyed in 64 days (see Figure 4).
c) An Epoxy resin (proprietary formulation)—End seal showed pronounced surface etching after one week and a ¼" deep conical cavity after 200 days immersion where the platinum anode protruded from the seal. The seal was still operative after this period, however (see Figure 5).
d) Polyblend (proprietary compound)—used as a vulcanized end seal for the anode. Slight surface chalking and an increase in hardness was noted after 60 days.
e) Neo-Polyblend (Portsmouth compound 50-C-1)—used as a vulcanized end seal for the anode. Surface attack but no appreciable damage was visible after 60 days.
f) Polyvinylchloride (cable sheath) surface turned brown in 14 days; no other defect was noted after 60 days.
g) Polyester potting compounds (various proprietary formulations—These were used as anode end seals. Very

Courtesy of Bureau of Ships

Figure 4. Platinum Anode with Polyester Grommet and Neoprene Cable Sheath after 63 days in Sea Water at 22 volts and 7½ Amperes.

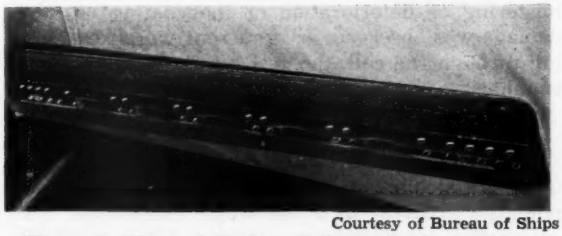
Courtesy of Bureau of Ships

Figure 5. Platinum Anode with Epoxy Grommet and Polythene Tube over Cable after 150 days in sea water at 22 volts and 7½ amperes.



Courtesy of Bureau of Ships

Figure 6. Boston Naval Shipyard glass reinforced polyester plastic holder for a platinum clad anode (unassembled).



Courtesy of Bureau of Ships

Figure 7. View of Boston platinum anode holder (assembled).

slight surface etching but no further effect was observed after 70 days immersion (see Figure 4).

h) Polyethylene (cable sheath)—No effect of any kind was observed in 140 days (see Figure 5).

It can be concluded from the above laboratory tests that the polyester and polyethylene resins were the most outstanding.

PLASTIC ANODE HOLDER ASSEMBLY

From the data obtained from the Portsmouth experiments, a new holder was designed using glass reinforced polyester as the material of construction. This choice of polyester over polyethylene or polyvinyl-chloride was dictated by its higher mechanical and impact strength and its thermosetting property which allows molding in the cold fluid condition. However, polyester-glass is somewhat higher in cost.

The design shown in Figures 6 & 7 was developed by Mr. B. H. Tytell at the Boston Naval Shipyard. You will note that the less expensive glass mat was used in the construction of the holder with no sacrifice of mechanical strength for the service intended. The plastic laminate conforms to the mechanical strength requirement of Specification Mil-P-17549, Grade D.

There are several features of this design which deserve mention. Note the scalloped edge and series of spaced holes along the length of the holder. The elimination of excess material in way of the anode reduces water resistance in way of anode thereby increasing current output. You will also note that the two part anode holder is symmetrical about the anode axis. This arrangement reduces fabrication cost and permits speedy dismantlement and assembly of the anode rod. The sections are bolted together with rigid PVC hardware. Provision has been made to coat the bottom flange of the holder with an epoxy resin coating for additional alkali resistance needed near the cathodic hull. The holder is attached by means of corrosion resistant bolts welded to the hull. The mounting hardware is then enclosed with a molded continuous housing of epoxy-glass laminate to prevent large current drain at the bolts. This measure increases the current distribution to remote areas of the hull where it is needed. The wide flange construction of the holder serves the additional purpose of providing an adequate dielectric shield for proper current distribution in way of the anode. The rubber mounting pad under the flange insures proper fit over minor irregularities in the hull and provides a measure of shock and vibration protection. The anode rod spacers are machined and may be cast from polyester-glass laminate or rigid PVC.

GRAPHITE ANODE HOLDER ASSEMBLIES

Previous designs of graphite anode holders varied from simple rod shapes clamped in rubber gasketed steel pipe clamps welded to the hull to elaborate streamlined steel boxes containing a graphite slab held in place by special insulated bolting arrangements. All these designs proved to be inadequate for many reasons, among which were insufficient shock and vibration protection, excess current drain at poorly insulated steel supports, and heavy, cumbersome mountings.

One design, utilizing a graphite slab, was fabricated from polyester-glass sheet materials. It consisted of a flat sheet plastic base to which were clamped plastic straps bridging across the anode at three locations. A neoprene gasket under each mounting strap held the anode securely to the base. This design is in service at present and although not ideal appears to be adequate. The major deficiencies are poor streamlining, and lack of protection against mechanical breakage.

A new graphite anode holder assembly has been designed at the Boston Chemical Laboratory which overcomes the objections of the previous designs. A view of the unassembled holder is shown in Figure 8. Again, as in the platinum-clad anode holder, glass reinforced polyester is the prime structural material. The glass-polyester can either be laid up over the graphite slab or a chopped glass polyester mix cast around it. The trapezoidal cross section of the graphite and the roughening of the embedded sides provide natural anchorage and water tight bonding within the plastic housing. The mounting studs and nuts are protected by a continuous cover and a rub-

Courtesy of Bureau of Ships

Figure 8. Boston Naval Shipyard polyester plastic housing for a graphite anode (unassembled).

Courtesy of Bureau of Ships

Figure 9. View of Boston Graphite anode holder (assembled).

ber blanket used as they were in the platinum-clad anode holder as shown in Figure 9. A comparatively large surface of graphite is exposed to give adequate current throwing power. The graphite is completely enclosed except for this conducting surface and will therefore provide protection against underwater shock and mechanical impact. An epoxy resin coating has been used on all areas which are prone to attack by the alkali cathodic reaction. An interesting detail in the new design is the use of an isocyanate foam material at the ends of the holder. This plastic is foamed in place where it bonds strongly to the polyester surface, prevents the entrance of water, and further cushions the anode slab against underwater shock or impact. Little is known of the electrochemical resistance of the isocyanates, but no contact with seawater is expected in this case because of complete enclosure by the polyester laminate. The cable connection is enclosed and is not in contact with the seawater.

A standard stuffing tube welded within the hull provides watertight penetration of the cable into the ship.

CATHODIC SHIELDS

From the introductory discussion we have learned that special dielectric shield materials are used in way of impressed current anodes mounted on a hull. These shields prevent the concentration of high current densities near the anode and therefore aid in spreading the current to more remote areas on the hull. High local current concentration is disruptive to most paint systems and therefore successful application of cathodic protection systems for shipboard use depend on quality shielding materials.

The requirements for suitable shield materials are as follows:

a) Resistance to alkali attack
b) Low permeability
c) Non-conductive
d) Adherent to steel and other metals
e) Easy to apply (preferably at room curing temperatures)
f) Resistant to impact, shock, vibration and flexing
g) Compatible with anti-fouling paints
h) Economical

With these objectives in mind several investigations were conducted concurrently. An accelerated test program was set up at several Government Laboratories and a long time immersion test was undertaken at the Harbor Island Test Station operated by the International Nickel Company.

BOSTON NAVAL SHIPYARD TESTS

Simple electrolytic cells were devised consisting of coated steel bar cathodes immersed in a seawater electrolyte and energized by graphite bar anodes. A potential of 8 volts was applied across each cell. Initially no current flowed through the cell because of the insulated coating on the cathode. Current could be measured as progressive deterioration occurred on the coated cathodes. Daily observations were made on each cell to check operating potential and current and to examine the specimens visually. If blistering or deterioration of the coating or other damage was observed, or if current was found flowing through the cell, the test was stopped. Table II summarizes the results of these tests.

PORTSMOUTH NAVAL SHIPYARD TESTS

Short term tests were conducted at Portsmouth using flat steel coated plates exposed under a 6 volt potential gradient in a sea water electrolyte. Two proprietary neoprene compounds 1/32″ in thickness and a vinyl plastisol 1/16″ in thickness were immersed for 21 days. The first neoprene compound which was a liquid application showed considerable ballooning indicating electro-osmotic transmission of water molecules through the coating. A similar thickness sheet neoprene specimen was unaffected. The same was observed for the vinyl plastisol. Other tests using 1.5 volt potentials were applied to cathodes coated with a vinyl plastisol over a vinyl primer, a polyester resin, liquid neoprene over neoprene primer, epoxy-polysulphide rubber blend, and an epoxy resin. These tests failed to disclose failures or to differentiate between the materials during a 21 day immersion period.

TABLE II—*Boston Naval Shipyard Tests*

Coating	Spec	Applied Pot.	No. Days	Remarks
Vinyl System	Round	8 volts	85 days	No current passing
	Flat	8 volts	85 days	excellent condition
Epoxy	Round	8 volts	85 days	No current passing
	Flat	8 volts	85 days	excellent condition
Epoxy 3-4	Round	8 volts	80 days	No current passing
	Flat	8 volts	80 days	excellent condition
Saran 113	Round	8 volts	8 days	Blistered, current passing
	Flat	8 volts	3 days	Blistered, current passing
Hot Sprayed Plastisol	Round	8 volts	60 days	No current passing good condition, slightly tacky
Polyester	Round	8 volts	30 days	No current flowing,
	Flat	8 volts	30 days	evidence of slight alkali attack
Heavy Pigmented Epoxy Based Cement	Flat	8 volts	2 days	Current flowing, blistered badly
Epoxy-glass reinforced			85 days	No current excellent condition

NEW YORK MATERIAL LABORATORY TESTS

A comprehensive program was undertaken at the New York Material Laboratory to determine alkali resistance and electro-osmotic properties of various

coating materials for ultimate use as dielectric shields. Two test criteria were used: alkali resistance to hot (135° F.) sodium hydroxide (4% NaOH) solution and resistance to a potential gradient of 6 volts. Various proprietary products were tested. Results of resistance to alkali immersion are shown in Table III.

TABLE III—*New York Material Laboratory Tests*
Appearance After Immersion in 4% Solution of Sodium Hydroxide

Material	Exposure Time	Appearance
Vinyl Formulation (18-25 Mils)	4 weeks	No visible deterioration
Vinyl Formulation (35-40 Mils)	4 weeks	No visible deterioration
Epoxy Formulation	4 weeks	No visible deterioration
Fluorinated Resin Formulation	1 week	½ of surface blushed
	2 weeks	¾ of surface blushed
	3 weeks	Large blister formed
Epoxy Formulation	4 weeks	No visible deterioration
Phenolic Formulation	4 weeks	No visible deterioration
Neoprene Formulation	1 week	Coating softened and blistered
	2 weeks	Blistering over larger area
	3 weeks	Blisters found to be filled with solution
Neoprene Sheet Stock	4 weeks	No visible deterioration
Nitrile Rubber Formulation	1 hour	Coating began to disintegrate immediately after immersion
Synthetic Rubber Formulation	1 hour	Coating began to disintegrate immediately after immersion
Synthetic Resin Formulation	4 weeks	Material softened & swelled
Epoxy Formulation	1 week	Slight blistering
	4 weeks	Slight blistering
Polyester Formulation	3 hours	Material began to disintegrate immediately after immersion
Vinyl System	4 weeks	No visible deterioration except for darkening of coating

Results of the resistance to an impressed voltage of 6 volts D.C. for periods up to 6 weeks show the following materials to be unsatisfactory:

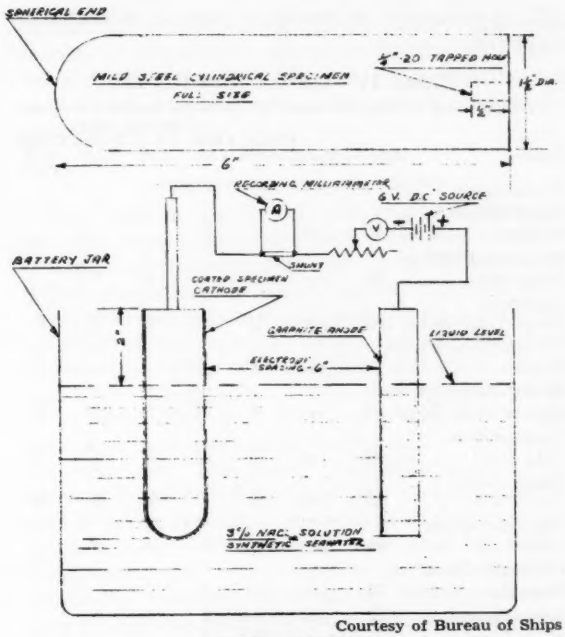
a) Vinyl paint formulation (18–25 mils)
b) Epoxy based coating (25–30 mils)
c) Fluorinated resin base coating (20–25 mils)
d) Modified thermosetting phenolic resin compound (30–40 mils)
e) Epoxy based coating (23–25 mils)
f) Brush on neoprene coating (25–30 mils)
g) Polyester resin (15–20 mils)

The only materials found satisfactory for both tests were as follows:

a) Vinyl paint formulation (35–40 mils)
b) Cold bond Neoprene sheet stock (1/16")
c) Navy vinyl paint system (25–30 mils)

The test set-up is illustrated in Figure 10.

To sum up, there are several materials that may be used as dielectric shields for cathodic protection systems. Costs of plastic materials compare favorably with the neoprene sheet stock and the cost of application is similar. Long term service evaluation is necessary to study the adhesion and durability of the plastics. Spraying or brushing a liquid to prepare a dielectric shield is more desirable than bonding or bolting sheet materials. There are several undesirable features of these tests which are overcome by long time immersion testing in open sea water. Accelerated tests can only give crude indications of what can be expected under specific conditions. For broad conclusive interpretation long term tests under reasonably comparable operating conditions are necessary. Further, there is evidence that intact coatings are more resistant to damage by electrolysis than scribed coatings of the same material.



Courtesy of Bureau of Ships

HALF SCALE

Figure 10. Test rig for determining electro-osmotic resistance of dielectric materials.

HARBOR ISLAND TESTS

Steel panels 12" x 12" x ⅛" were prepared in triplicate with many suitable coatings. One panel was retained as a control and the other two were exposed to specific potential gradients. Panels were sandblasted and degreased prior to application of the coatings. Some were scribed to bare metal, others left intact. The panels were mounted on insulated racks and immersed below mean low tide at Wrightsville Beach, N.C. The panels were examined at four month intervals. Applied potentials were started at 1.5 volts and were increased in increments of 1.5 volts after each inspection interval. All coating failures were removed at each inspection and those panels which appeared to be effective were re-immersed and the next higher potential applied. The coated panels were painted on half the area of one side with one coat of Navy Formula 117 and over entire face area with two coats of vinyl anti-fouling paint, Navy Formula

121. A summary of results to date is tabulated in Table IV.

TABLE IV—*Harbor Island Tests*

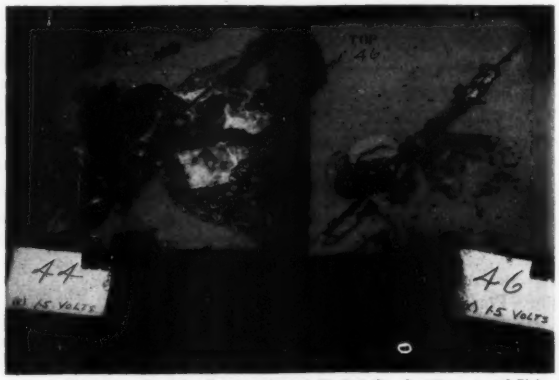
U—Unaffected; S—Slight Damage; M—Moderate Damage; F—Failed

Coating	Mils	Scribed	TOTAL TIME 15 mos. 4 mos. Control	1.5V	7 mos. 3.0V	15 mos. 4.5V	15 mos. A. F. Paint
Flame Spray Polysulphide rubber	45	yes	S	F	..	..	..
Neoprene Brush on Heat cure	50	yes	M	U	S	F	..
Neoprene Sheet Cold Bond	100	yes	U	U	U	U	F
Polyethylene Flame Spray	50	yes	M	F	..	..	..
Epoxy-Polysulphide Rubber Cold Bond	90	yes	S	S	S	M	F
Epoxy-Glass Cold Cure	65	yes	U	U	S	S	M
Epoxy-Glass Heat Cure	65	yes	S	U	U	S	M
Polyester-Glass Cold Cure	65	yes	U	S	M	F	..
Polyester-Glass Heat Cure	70	yes	U	S	F	..	..
Navy Saran	23	yes	M	F	..	..	..
Navy Vinyl	21	yes	U	F	..	..	..

The conclusions from all these tests indicate thus far that cold bond neoprene sheet rubber ⅛″ thickness and cold cure epoxy-glass resins 1/16″ thickness show outstanding performance as dielectric shields. Vinyl films over 40 mils show promise and need further investigation. Heat cure systems are not considered practical for direct application to a ship's hull. Service tests performed with cold bond neoprene confirm the existing experimental data. Further service tests are needed for epoxy-glass laminates as dielectric shields. The cost of materials and application compare favorably with each other at the present time. Figures 11 through 14 show typical failures of certain materials after 4 months immersion at cathode potentials of 1.5 volts.

Photographed by International Nickel Co. for Bureau of Ships

Figure 11. Flame sprayed polysulphide rubber (40 mils) appearance of scribed cathode after 4 months immersion at 1.5 volts applied potential.

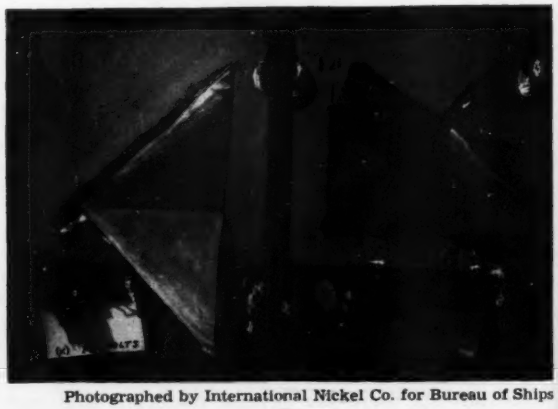


Photographed by International Nickel Co. for Bureau of Ships

Figure 12. Flame sprayed Polyethylene (50 mils)—panel 44 appearance of scribed coated cathode after 4 months immersion at 1.5 volts applied potential.

Photographed by International Nickel Co. for Bureau of Ships

Figure 13. Brush coated Navy saran anti-corrosive and anti-fouling paint system (23 mils) appearance of scribed coated cathode after 4 months immersion at 1.5 volts applied potential.



Photographed by International Nickel Co. for Bureau of Ships

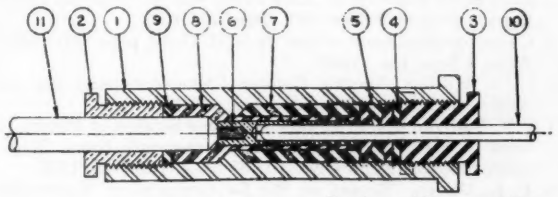
Figure 14. Brush coated Navy vinyl paint system (21 mils) appearance of scribed coated cathode after 4 months immersion at 1.5 volts applied potential.

CABLE SEALS

Cable seals have been the weakest link in impressed current cathodic protection systems. They perform a two-fold vital function of making a watertight electrical connection between a platinum-clad anode and the current carrying cable lead and provide a watertight penetration through the underbody of the ship. The seal must not only adhere to the cable sheath and anode rod but it must also end seal the cable insulation and bare conductors. Molded rubber seals both for submarine and surface ship service have been in existence for many years. When tried on platinum-clad rods in actual service, the conventional seals proved inadequate for two reasons:

a) Lack of adhesion to the platinum-clad rod
b) Oxy-chloride attack of the seal

Portsmouth Naval Shipyard undertook the design of a molded cable seal which would adhere to platinum and developed a successful technique using special rubber formulation.[8] These seals withstood high pressures of several hundred psi. Although the seal was satisfactory from a waterproof standpoint, previous experience indicated that certain neoprene swelled and embrittled when exposed to inert anode reaction products. Therefore, a special stuffing tube was fabricated to enclose the approved seal. Special "Teflon" packing rings were used over the seal at the anode end to prevent oxychlorides from coming in contact with the rubber. This type seal has of late been redesigned to provide easy assembly and incorporates a polyester gland nut at the anode end for protection against excess current drain. The cross section of this seal is shown in Figure 15. Future work is going ahead to discover proper potting precedures for suitable resins which could replace the complicated molded rubber seal.



Courtesy of Bureau of Ships

LIST OF MATERIAL

Pc. No.	Name	Qty.	Material
1	Body	1	Steel
2	Nut. Gland size 'B'	1	
3	Nut. Gland	1	Polyster Resin Glass Reinforced
4	Packing Disc	1	Teflon
5	Cup & Cone Packing	1	Teflon
6	Connector	1	Phos. Bronze
7	Grommet	1	Rubber
8	Molded Packing	1	Rubber
9	Ring, Gland size 'B'	1	
10	Anode ¼' dia. (for info. only)	..	Platinum Clad Silver
11	Cable—SHOF 23	As Reqd.	Polychloroprene Sheathed
12	Compound, 50.D-1	As Reqd.	

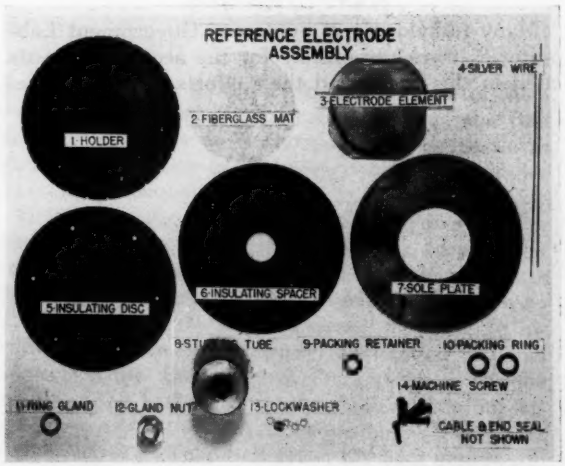
Figure 15. Portsmouth Naval Shipyard design of a pressure proof stuffing tube and seal for platinum clad anode rod.

REFERENCE ELECTRODE HOLDER

One other component of cathodic protection system that deserves mention is the reference electrode holder. The reference electrode assembly has a simplified streamlined housing molded for rigid polyvinyl chloride. This material was chosen for its good molding characteristics, high impact resistance, chemical inertness and excellent dielectric properties. The assembly has a sole plate cut from PVC sheet stock. The mold is so made that water flood holes and slots are cast in the top half of housing in one operation. The silver-silver chloride electrode element is cushioned by glass fiber mat and extreme care is taken to make a waterproof cable connection to the element. Plastic resins are used for sealing and insulating all exposed metal parts which are in proximity with the silver-silver chloride element. Any grounded bare metal would affect the operation of the instrument. The only disadvantage encountered with this design is the high initial cost encountered in the fabrication of steam jacketed molds. Figure 16 shows the various pieces used in the construction of the reference electrode. And Figure 17 shows the assembled electrode.

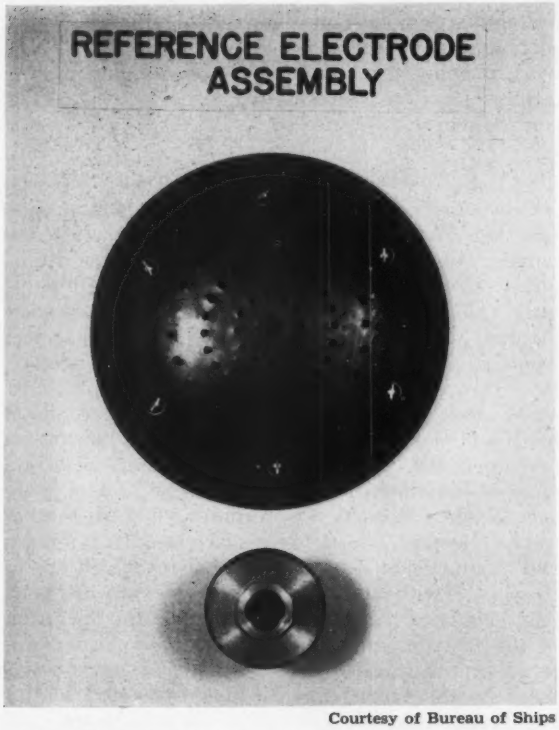
CONCLUSION

Plastics have proved to be versatile materials with almost an unlimited range of properties and are admirably suited for use in fabrication of cathodic protection equipment. No doubt, as the art progresses, streamlining and simplified construction will enlarge the scope of plastics in this field. New formulations



Courtesy of Bureau of Ships

Figure 16. Unassembled Portsmouth Naval Shipyard silver-silver chloride reference electrode featuring a rigid polyvinyl chloride molded housing.



Courtesy of Bureau of Ships

Figure 17. Assembled view of Portsmouth Naval Shipyard reference electrode.

for obtaining special chemical or electrical properties are on the horizon. As they become available commercially, they will no doubt be incorporated into new designs.

Plastics technology has caused significant advances in all branches of engineering—cathodic protection is indeed one of them.

ACKNOWLEDGEMENTS

Many individuals at numerous Government Laboratories were responsible for the accomplishments cited in this paper and their efforts are hereby acknowledged. In particular, special mention is made to the following persons:

1. Bureau of Ships—Messrs. F. E. Cook & W. Graner for their general contributions to the overall subject.
2. Boston Naval Shipyard—Mr. B. H. Tytell for his work on platinum-clad and graphite anode holders.
3. Harbor Island Test Station—Mr. R. B. Teel for his work on long time immersion tests on cathodic shields.
4. Naval Research Laboratory—Mr. E. E. Nelson for his early work on the development of anode holders and reference electrode assemblies for cathodic protection applications.
5. New York Material Laboratory—Messrs. S. Weiss and A. W. Cizek, Jr., for their work on accelerated electro-osmosis experiments.
6. Norfolk Naval Shipyard—Mr. W. J. Francis for his work on long time immersion tests on cathodic shields at Harbor Island.
7. Portsmouth Naval Shipyard—Mr. F. L. Downs for his work on cable seals and anode.
8. Resistant Materials—Mr. E. C. Taylor and staff for their efforts on the manufacturing reference electrode assembly and Mr. R. F. Hayes and staff for their many excellent design drawings.

The opinions expressed in this article are those of the authors and not necessarily those of the Navy Department.

BIBLIOGRAPHY

1. F. L. Laque, "Behavior of Metal and Alloys in Sea Water," article in *Corrosion Handbook*, edited by H. H. Uhlig, page 313, John Wiley & Sons, Inc. (1948).
2. U. S. Patent No. 2,200,469 (May 14, 1940), G. C. Cox, "Anti-corrosive and Anti-fouling Coating and Method of Application."
3. F. E. William, *Smith's College Chemistry*, page 520, D. Appleton-Century Company, Inc. (1946).
4. Transportation Board, Fort Eustis, Virginia, "Electrical Process for Removal of Scale From Vessels," page 41, Final Report, Project 9-99-01-001 (May 12, 1952).
5. *Corrosion Handbook*, edited by H. H. Uhlig, page 1137, John Wiley & Sons, Inc. (1948).
6. F. E. William, *Smith's College Chemistry*, page 154, D. Appleton-Century Company, Inc. (1946).
7. J. F. Sullivan and F. L. Downs, "Some Studies of Materials Used in Platinum Anode Seals," Portsmouth Naval Shipyard, Progress Report No. RD 55-95LF (May 20, 1955).
8. F. L. Downs, "Report on the Development of Watertight Seals on Platinum Clad Anodes for Submarine Hulls," Portsmouth Naval Shipyard, Technical Report No. RD 53-30CF (Final) (December 11, 1953).

WILLIAM J. ROBERTS and IRVING GRANET

THE VARIATION OF TEMPERATURE IN FIXED FUEL ELEMENTS OF A NUCLEAR REACTOR UNDER LOAD AND COOLANT-TEMPERATURE CHANGES

IRVING GRANET

was born July 2, 1924. Attended Brooklyn Technical High School and received Bachelor of Mechanical Engineering Degree from Cooper Union in 1944. Served two years with Air Corps overseas as Flight Engineer. Employed 1944-45 on Navy Torpedo project as project engineer on stress and vibration studies. Since 1947, employed as project engineer, Research Department, Foster Wheeler Corporation. In June 1948 received Master of Mechanical Engineering Degree from Polytechnic Institute of Brooklyn and in 1949 received Professional Engineering License in New York. At present going to Brooklyn Polytechnic for Doctorate degree and has taught thermodynamics in the evening session. Has published several articles on properties of gases, thermodynamics and applied mechanics. Present duties are as assistant to the Contract Manager of the Nuclear Energy Department.

WILLIAM J. ROBERTS

was born March 5, 1928. He attended Alabama Polytechnic Institute, receiving a B.S. in chemical engineering in 1947 and an M.S. in Theoretical Physics in 1949. From 1949 to 1951 he was a Research Assistant in Physics at Pennsylvania State while pursuing studies toward his Doctorate in Physics. From 1951 to 1954 he served in the Navy as a Lieutenant (junior grade) with duty as Ship Superintendent (submarines), Philadelphia Naval Shipyard. In 1955 he joined Union Carbide and Carbon at Oak Ridge as Associate Physicist in the Theoretical Physics Division of K-25. He recently attended the Oak Ridge School of Reactor Technology.

THE RECENT ACTION by the United States Congress which provided funds for the construction of nuclear powered naval surface units and the request of President Eisenhower for the construction of a nuclear powered merchant vessel have both focused attention on the use of nuclear reactors for surface vessel propulsion. The reactor on which the most experience is available and which has been perfected to the largest extent is the pressurized water type. Such a reactor, which is water cooled, water moderated and has enriched uranium fuel plates, now powers the *Nautilus* and is being built for electric power generation at Shippingport, Pennsylvania (1). It must naturally be assumed that a pressurized water reactor is being seriously considered for surface vessel application. Since one of the major problems involved in naval applications is that the reactor may be called upon to respond to severe load changes and one of the design limitations of such reactors is the maximum temperature of the fuel plates, it becomes both necessary and desirable to study the variation of fuel plate temperature under transient conditions.

Under operating conditions the heat generation in the fuel plates of this type of reactor is, for all engineering purposes, a function of the thermal flux distribution. As indicated by Glasstone (1), one may idealize the plate type fuel element as being an infinite slab with a uniformly distributed thermal source. While the thermal flux varies with the length of the fuel element, we shall confine our attention to a given section of the element. Thus, across a given section of the fuel plate, the heat generation will be only a function of time and the "x" coordinates as shown in figure 1. It will be assumed that the physical properties of the fuel plate, i.e. thermal

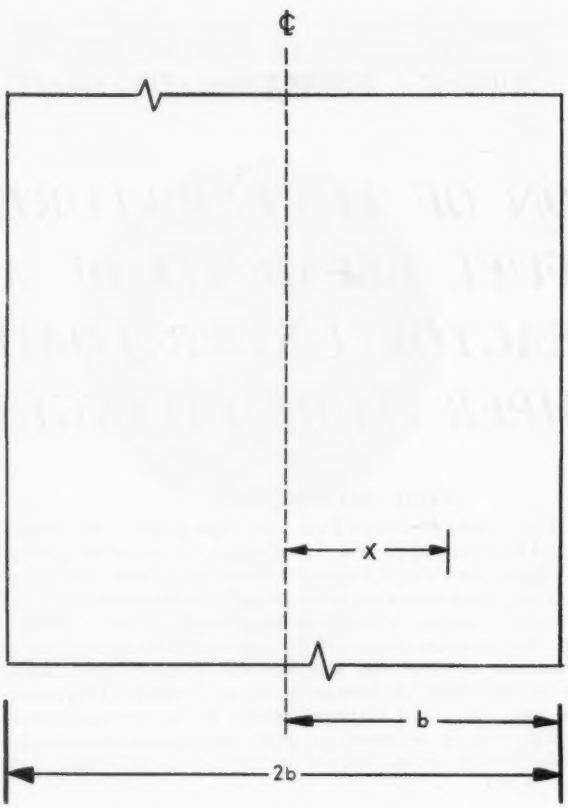


Fig. 1. Typical Dimensions of Fuel Plate

conductivity, specific heat and density, are independent of time, space and temperature.

In order to further simplify the analysis two restrictions will be placed upon the problem. The first of these is that the heat generation varies linearly with time. This restriction is not too limiting since any non-linear variation may be approached by successively superposing linear variations. The second restriction that will be made is that the surface temperature of the element varies linearly with time. Such an assumption is more limiting than the restriction placed upon the variation of heat generation. However, the heat transfer coefficient is usually quite high in these reactors and the error may not be too serious. For those cases where the heat transfer coefficient is low or highly variable, it may be profitable to use graphical or numerical methods (2).

On the basis of the above discussion, the general time dependent heat transfer equation may be written as equation 1. In the notation used, the subscripts refer respectively to space and time.

$$k\nabla^2 T_{(x,t)}+G_{(t)}=\rho c_p\frac{\delta T_{(x,t)}}{\delta t} \quad \ldots\ldots\ldots\ldots\ldots (1)$$

Since it has been postulated that both the generation rate and the surface temperature of an element vary linearly with time, the functional relations of these variables are given in equations 2 and 3. A positive value of "a" denotes an increase of generation rate with time and conversely, a negative value would denote a decrease of generation rate with time. However, a positive value of "c" denotes a negative alteration of surface temperature with time while a negative value denotes an increase of surface temperature with time.

$$G=G_o(1+at) \quad \ldots\ldots\ldots\ldots\ldots (2)$$

$$\left.\frac{\delta T}{\delta t}\right|_{x=b}=-c \quad \ldots\ldots\ldots\ldots\ldots (3)$$

Equation 1 is of the form that a general solution will consist of the sum of the time independent part plus the time dependent part. Therefore, a solution of the form of equation 4 is postulated as a solution to equation 1. In equation 4, the first term on the right hand side is defined by equation 5.

$$T_{(x,t)}=T_o+T_{1(x,t)}+T_{2(x,t)}+T_{3(x,t)}=-ct \ldots. (4)$$

$$T_{bo}=T_o=\text{arbitrary base temperature} \quad \ldots\ldots (5)$$

In order that equation 4 be a solution to equation 1 it is necessary to properly select the functional relations for T_1, T_2, and T_3. Equations 6 through 11 give the required relations:

$$k\nabla^2 T_1+G_o=O \quad \ldots\ldots\ldots\ldots\ldots (6)$$

$$k\nabla^2 T_2+aG_o t=\rho c_p\frac{\delta T_2}{\delta t} \quad \ldots\ldots\ldots\ldots\ldots (7)$$

$$T_{2(x,o)}=O \quad \ldots\ldots\ldots\ldots\ldots (8)$$

$$k\nabla^2 T_3=\rho c_p\frac{\delta T_3}{\delta t}-c \quad \ldots\ldots\ldots\ldots\ldots (9)$$

$$T_{3(x,o)}=O \quad \ldots\ldots\ldots\ldots\ldots (10)$$

$$\left.\frac{\delta T_1}{\delta t}\right|_{x=b}=\left.\frac{\delta T_2}{\delta t}\right|_{x=b}=\left.\frac{\delta T_3}{\delta t}\right|_{x=b}=O \quad \ldots\ldots\ldots\ldots\ldots (11)$$

Physically, equation 4 represents time dependent temperature profiles based on a moving basal temperature which decreases (or increases) at a rate of "c" degrees per second. One may further interpret the terms in equation 4 as follows:

T_o = base temperature at t=O
T_1 = parabolic temperature profile due to G_o (Ref. 1)
T_2 = change in T_1 due to increase in G
T_3 = variation due to decrease in surface temperature
$-ct$ = moving basal temperature
} .... (12)

The solution of equation 7 proceeds as follows: introducing the new variables of equations 13 into equation 7 one obtains equation 14.

$$\left.\begin{aligned} y&=\left(\frac{\pi x}{2b}\right)\\ v&=\frac{K\pi^2}{4\rho c_p b^2}t\\ w&=\frac{aG_o}{\rho c_p}\end{aligned}\right\} \quad \ldots\ldots\ldots\ldots\ldots (13)$$

$$\frac{\delta^2 T_2}{\delta y^2}+v\,w=\frac{\delta T_2}{\delta v} \quad \ldots\ldots\ldots\ldots\ldots (14)$$

A solution to equation 14 can be arrived at by several approaches. In this case it is felt that the use of operational calculus considerably shortens the labor necessary to obtain the desired solution. Using the Laplace transform (3), the procedure is as follows:

$$L(T_2)=\int_o^\infty T_2 e^{-sv}\,dv=\Theta \quad \ldots\ldots\ldots\ldots\ldots (15)$$

and equation 14 becomes:

$$\frac{\delta^2\Theta}{\delta y^2}+\frac{w}{S^2}=S\Theta \quad \ldots\ldots (16),\ (\text{since } T_{2(x,o)}=O)$$

Θ may now be expanded in a Fourier cosine series (3) in the interval from $\frac{-\pi}{2}$ to $\frac{\pi}{2}$.

$$\Theta=\sum_n d_n \cos ny \quad \ldots\ldots\ldots\ldots\ldots (17)$$

Substitution of equation 17 into equation 16 yields:

$$d_n(n^2+s)\cos ny=\frac{w}{s^2} \quad \ldots\ldots\ldots\ldots\ldots (18)$$

To evaluate the coefficients d_n the orthogonal properties of trigonometric functions are utilized. Both sides of 18 are multiplied by cos my dy and integrated from O to $\frac{\pi}{2}$.

$$d_n(n^2+s)\frac{\pi}{4}=\frac{w}{ns^2}(-1)^{\frac{n-1}{2}} \quad \ldots (n\text{ odd}) \ldots (19)$$

Thus equation 17 becomes

$$\Theta_{(x,s)}=\frac{4w}{\pi}\sum_{n\ odd}(-1)_n^{\frac{n-1}{2}}\left(\frac{\cos ny}{s^2(n^2+s)}\right) \quad \ldots\ldots\ldots (20)$$

Taking the inverse transform and replacing the original variables, the desired result for $T_{2(x,\,t)}$ is obtained:

$$T_{2(x,t)}=\frac{4aG_o}{\pi\rho c_p}\sum_{n\ odd}\frac{(-1)^{\frac{n-1}{2}}}{n^5}\cos\left(\frac{n\pi x}{2b}\right)\left\{e^{\frac{-n^2k\pi^2}{4b^2\rho c_p}}+\frac{n^2\pi^2k}{4b^2\rho c_p}t-1\right\} \quad \ldots (21)$$

In order to complete the general solution it only remains to evaluate equation 9. Retaining the definitions of y and v from equation 13, we define a new variable in equation 22 in place of the variable w.

$$B=\frac{4\rho c_p b^2 c}{\pi^2 k} \quad \ldots\ldots\ldots\ldots\ldots (22)$$

The procedure at this point is the same as the one utilized in the evaluation of equation 7. Equation 22 is substituted into equation 9 to yield equation 23.

$$\nabla^2 T_3=\frac{\delta T_3}{\delta v}-B \quad \ldots\ldots\ldots\ldots\ldots (23)$$

Utilizing the Laplace transform, one arrives at equations 24 and 25.

$$\Theta_3=\int_o^\infty T_3 e^{-sv}\,dv \quad \ldots\ldots\ldots\ldots\ldots (24)$$

$$\nabla^2\Theta_3-s\Theta_3=-\frac{B}{s} \quad \ldots\ldots\ldots\ldots\ldots (25)$$

In arriving at equation 25, use has been made of the condition of equation 10. As before, Θ_3 is expanded in a Fourier cosine series.

$$\Theta_3=\sum_n D_n \cos ny \quad \ldots\ldots\ldots\ldots\ldots (26)$$

$$\sum_n D_n(n^2+s)\cos ny=\frac{B}{s} \quad \ldots\ldots\ldots\ldots\ldots (27)$$

The coefficients D_n are evaluated by utilizing the orthogonal properties of trigonometric series, yielding equation 28:

$$D_n=\frac{4B}{\pi ns}\frac{(-1)^{\frac{n-1}{2}}}{(s+n^2)} \quad \ldots\ldots (\text{for n odd}) \ldots\ldots\ldots (28)$$

and therefore:

$$\Theta_3=\frac{4B}{\pi}\sum_{\text{"n" odd}}\left(\frac{(-1)^{\frac{n-1}{2}}}{n}\right)\frac{\cos ny}{s(s+n^2)} \quad \ldots\ldots (29)$$

Taking the inverse transform of equation 29 and reintroducing the original variables of the problem, the solution for $T_{3(x,\,t)}$ may be written as:

$${}_3T_{(x,t)}=\frac{16\rho c_p b^2 c}{\pi^3 k}\sum_{n\ odd}\frac{(-1)^{\frac{n-1}{2}}}{n^3}\cos\left(\frac{n\pi x}{2b}\right)\left\{1-e^{-\frac{n^2\pi^2 k}{4b^3\rho c_p}t}\right\} \quad \ldots\ldots\ldots\ldots (30)$$

The complete solution to equation 1 may now be written as the term by term sum of equation 4. Performing this operation yields equation 31, which is the complete solution. For other than linear variations in either the heat generation rate or the surface temperature, the analysis proceeds as above. However, the mathematical labor may be excessive before a solution can be obtained.

$$T_{(x,t)}=T_o+\frac{G_o}{2K}(b^2-x^2)+\frac{4aG_o}{\pi\rho c}\sum_{n\,odd}\frac{(-1)^{\frac{n-1}{2}}}{n^5}\cos\left(\frac{n\pi x}{2b}\right)\left\{e^{-\frac{n^2\pi^2k}{4b^2\rho c_p}t}+\frac{n^2\pi^2k}{4b^2\rho c_p}t-1\right\}+\frac{16\,\rho c_p b^2 c}{\pi^3 k}\sum_{n\,odd}\frac{(-1)^{\frac{n-1}{2}}}{n^3}\cos\left(\frac{n\pi x}{2b}\right)\left\{1-e^{-\frac{n^2\pi^2k}{4b^2\rho c_p}t}\right\}-ct \quad (31)$$

NOMENCLATURE

k = Thermal conductivity BTU per hour per square ft. per degree F per foot
x = Position coordinate—ft.
t = time—hrs.
T = Temperature °F
T_o = Arbitrary base temperature °F
G = Generation rate BTU per cubic foot per hour at any time t
G_o = Generation rate BTU per cubic foot per hour prior to change (steady state)
a = Coefficient of generation rate change—per hour
c = Coefficient of temperature change $= \left.\frac{-\delta T}{\delta t}\right|_{x=b}$
T_1, T_2, T_3 = Temperature functions °F
b = Half width of element—ft.
$y=\frac{\pi x}{2b}$
$v=\frac{K\pi^2 t}{4\rho c_p b^2}$
$w=\frac{aG_o}{\rho c_p}$
s = Variable of Laplace transform
n = integer
n odd = odd integers 1, 3, 5, 7 . . . etc.
$e = 2.71828\ldots$
$B=\frac{4\rho c_p b^2 c}{\pi^2 k}$
subscript (x,t) = position and time respectively
∇^2 = Laplacian (Ref. 1, 3)
ρ = Density—pounds per cubic ft.
c_p = Specific heat BTU per pound per degree F
$\pi = 3.1416\ldots$
$\Theta = L(t) =$ Laplace transform of $(t) = \int_o^\infty te^{-sv}\,dv$
D_n and d_n = Coefficients of Fourier Cosine Series
Σ = Summation over the range of the variable

REFERENCES

1. Principles of Nuclear Reactor Engineering by S. Glasstone, D. Van Nostrand Co., 1955.
2. Shortcuts for Transient Heat Flow by I. Granet and R. M. Gould, Chemical Engineering, Feb. 1956, pp. 183-186.
3. Mathematical Methods in Engineering by Theo. von Karman and M. A. Biot, McGraw-Hill Book Co., 1940.

USE OF SODIUM AND OF SODIUM-POTASSIUM ALLOY AS A HEAT-TRANSFER MEDIUM

ACKNOWLEDGEMENT

This is an abridged version of a paper read before the Institution of Mechanical Engineers on January 13, 1956, and published in the January 20 and 27 editions of "The Engineer." The authors are W. B. Hall, Principal Scientific Officer of the Research and Development Branch of the United Kingdom Atomic Energy Authority, and T. I. M. Crofts, also of the United Kingdom Atomic Energy Authority.

THE ADVENT of the nuclear reactor as a heat source, with its almost unlimited potential heat rating, has stimulated the search for a heat-transfer medium capable of operating over a wide temperature range with a good heat-transfer coefficient. As a consequence, the use of liquid metals for this purpose has received much attention during the past few years. In the design of a reactor cooling circuit, many limitations are imposed by nuclear physics, and it is not necessarily the case that a good reactor coolant is also the best choice for an industrial heat-transfer process. It appears, however, that sodium and sodium-potassium alloy are suitable for general application to heat-transfer problems when high heat fluxes and temperatures are concerned.

For industrial application the wide temperture range over which many liquid metals can be used is likely to be at least as attractive as the high heat-transfer coefficients which are obtained. In contrast with the nuclear reactor, it may be neither necessary nor desirable to operate at heat fluxes which are as much as ten times those in a modern water-tube boiler; much of the apparent advantage of so doing is lost in excessive temperature drops through walls of pipes. For example, it is possible to achieve a heat flux of 1kW per square centimeter (about 3×10^6 B.T.U. per square foot per hour) with sodium at a temperature difference between the surface and the liquid metal of about 100 deg. Cent.; at this heat flux the temperature gradient in a stainless steel wall would be about 500 deg. Cent. per millimeter (or 250 deg. Cent. across a 0.020in wall). Clearly, if the temperature drop across the wall is not to dominate the heat-transfer process the wall must be exceedingly thin; conversely, if the wall cannot be thin, such high heat fluxes will produce excessive thermal stresses in it.

At the outset of the work described in this paper the choice of liquid metal seemed to be between mercury, sodium and sodium-potassium alloy, and the various lead-bismuth or lead-bismuth-tin alloys. It was clear from the literature on the subject that sodium or sodium-potassium had an advantage over mercury and lead-bismuth as far as corrosion of steel was concerned; it is also cheaper, and it has a much more useful working temperature range than mercury. Mercury has the further disadvantage that its vapor is highly toxic. The main disadvantages of the alkali metal lie in the prevention of oxidation either inside the apparatus where the oxide may block the pipes, or outside where it may cause a fire; there is also the reaction with water, which may in certain circumstances cause an explosion. In spite of these disadvantages it was decided that the work should be concentrated upon sodium and sodium-potassium alloy.

An important factor enabling this decision to be reached was the pioneer work carried out by a group at the Atomic Energy Research Establishment at Harwell under the leadership of Dr. S. Bauer; some

of this work has been reported in the *Proceedings* of the Liquid Metal Utilization Conference, 1954.

The initial experimental program was directed mainly along two lines: first, measurements of heat-transfer coefficients under conditions of cleanliness which could reasonably be attained in an industrial plant; secondly, the investigation of handling problems, such as pumping, flow measurement, level control, and the cleaning of plant before and after its use with the liquid metal. The work was later extended to cover cavitation, the testing of heat exchangers for use with sodium on one side and water on the other, the calibration of electromagnetic flow meters, and the investigation of methods of measuring and controlling the amount of oxide present in the apparatus. An important by-product of the work is the experience gained both in the design and the operation of liquid metal circuits. Throughout the work emphasis has been placed on the use of conventional techniques wherever possible; this principle has been departed from only where the use of a liquid metal introduces a simpler process, or where experience has shown normal practice to be unsuitable.

TABLE I—*Notations*

- c—Specific heat of liquid metal, cal. gram^{-1} deg. Cent.$^{-1}$
- d—Diameter of tube, centimeters.
- d_e—Effective diameter of annulus, $d_o - d_i$, centimeters.
- d_i—Inner diameter of annulus, centimeters.
- d_o—Outer diameter of annulus, centimeters.
- h—Differential pressure across Venturi, kilograms centimeters^{-2}.
- k—Thermal conductivity of liquid metal, cal. centimeters^{-1}, sec.$^{-1}$, deg. Cent.$^{-1}$
- P_c—Cavitation pressure, kilograms centimeters^{-2}.
- P_i—Pressure at pump inlet, kilograms centimeters^{-2}.
- Pr—Prandtl number, $c\mu/k$, dimensionless.
- P_1—Pressure at Venturi inlet, kilograms centimeters^{-2}.
- Nu_a—Nusselt number for an annulus, $\propto d_e/k$, dimensionless.
- Nu_t—Nusselt number for a tube, $\propto d/k$, dimensionless.
- r_i—Inner radius of annulus, centimeters.
- r_o—Outer radius of annulus, centimeters.
- Re—Reynolds number, $Vd\rho/\mu$, dimensionless.
- V—Velocity, centimeter second^{-1}.
- $\propto$—Heat-transfer coefficient, cal. centimeters^{-2}, second^{-1}, deg. Cent.$^{-1}$
- ρ—Density of liquid metal, gram centimeters^{-3}.
- μ—Viscosity of liquid metal, gram centimeters^{-1}, second^{-1}.

DESCRIPTION OF EXPERIMENTAL PLANT

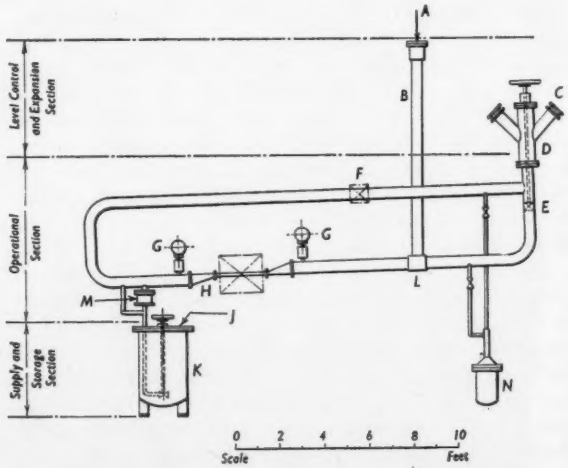
Eight liquid metal circuits have been built and operated during the three years in which the work has been in progress. Most of these conform to one of the three types which are described below. Detailed discussion of the features common to all the circuits and of the methods employed in their construction is given under the heading "Design and Operational Problems."

Electromagnetic Pump Test Circuit

A diagram of the equipment is shown in Fig. 1. Its purpose was to measure the performance of an electromagnetic induction pump over a range of temperatures and with sodium-potassium alloys of various compositions. The pump was placed in a rectangular loop in stainless steel pipe of 4in nominal bore; after leaving the pump the metal passed through a flow meter with a nozzle 3.15in. in diameter, a control valve, an electromagnetic flow meter, and thence back to the pump inlet. Measurements of the inlet and outlet pressures at the pump and the differential pressure across the nozzle flow meter enabled the characteristic curve of the pump to be determined.

The lowest point of the main circuit was connected to a reservoir tank from which the liquid metal could be displaced into the circuit by nitrogen under pressure. This tank was fitted with three globe valves, which were immersed in the liquid metal and operated by spindles passing through glands in the top of the tank. The first valve was used for filling the tank with sodium-potassium alloy from the steel carboys in which it is supplied. The second valve was used to fill the circuit through a sintered stainless steel filter, and the third to drain the circuit, bypassing the filter. The tank was fitted with a level measuring device, which is described in the section on "Level Measurement."

The control valve, situated at the highest point of the circuit, was also employed as an expansion chamber, and the valve spindle passed through the nitrogen-filled space above the liquid metal surface



A—Nitrogen supply.
B—Manometer tubes
C—Inspection ports.
D—Expansion and level control chambers.
E—Control valve.
F—Electromagnetic flow meter.
G—Diagram pressure gauges with trapped gas vessels.
H—Electromagnetic pump.
J—Nitrogen supply.
K—Reservoir tank.
L—I.S.A. nozzle flow meter.
M—Filter.
N—Cold trap.

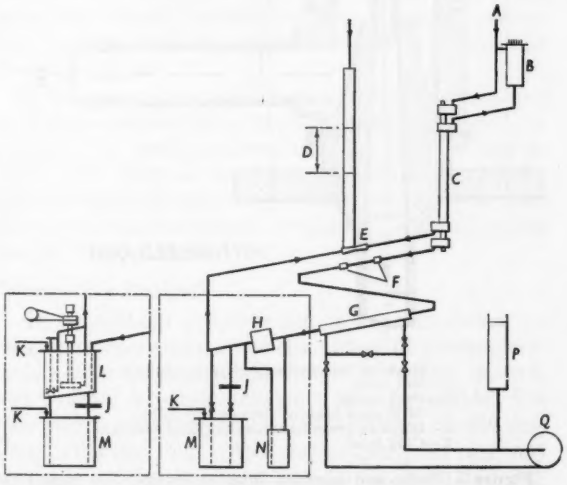
Figure 1. Layout of electromagnetic pump circuit.

and then through a gland. At a later stage observation windows were fitted above the metal surface.

Finally, the circuit was fitted with a by-pass cold trap, for the continuous removal of oxide from the liquid metal; this extremely important device is fully described under the heading "Oxide Removal." Initial tests were terminated by oxide contamination after only a few hundred hours; since the cold trap was fitted the circuit has run for 10,000 hours without contamination.

Apparatus for Heat-Transfer Measurements

The apparatus, which is illustrated in Fig. 2, is of the "figure-of-eight" type, in which the heat exchanger under test is used regeneratively, thus reducing the amount of heat which must be supplied for a given total heat exchange. It was decided to use a double-annulus heat exchanger (Fig. 3) rather than the more usual tube and annulus design, since it would then be possible to measure axial temperature gradients on the inner as well as the outer tube, and thus to separate the effect of the entrance on the heat-transfer coefficient. The flow of liquid metal was maintained by a centrifugal pump in the earlier experiments, and later by a single phase electromagnetic pump. The centrifugal pump was immersed in the liquid metal and driven by a shaft, carried on bearings outside the tank, which passed through a gas-seal gland in the tank lid. Some trouble was experienced owing to the presence of sodium vapor in the gas space, and maintenance of a good gas seal was difficult; no doubt this could have been overcome by further development work, but at this time a small single phase electromagnetic pump became



A—Nitrogen supply.
B—Electric heater tank.
C—Heat exchanger (see Fig. 3).
D—Pressure difference across Venturi meters.
E—Venturi meter.
F—Electromagnetic flow meter.
G—Cooler.
H—Electromagnetic pump.
J—Filters.
K—Nitrogen supply.
L—Mechanical pump.
M—Reservoir pump.
N—Cold trap.
P—Coolant reservoir.
Q—Coolant circulating pump.

Figure 2. Layout of heat-transfer apparatus—Showing mechanical pump in early experiments with sodium and electromagnetic pump in experiments with sodium-potassium.

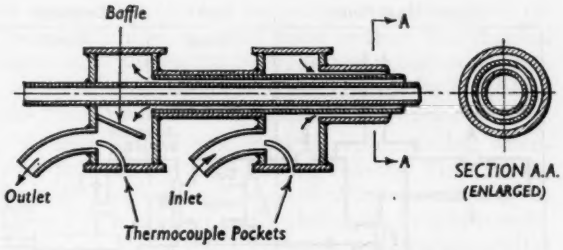


Figure 3. Detail of double-annulus heat-exchanger end.

available, and was therefore used to replace the centrifugal pump. Flow measurement was by means of a Venturi tube, the pressure difference being measured by means of an inverted U-tube, as will be described in the section "Level Measurement."

A heat input of 20kW could be effected by means of electric heaters at one end of the loop, and a sodium to Downtherm heat exchanger was used to reject this heat at the other end. Measurements of the temperature rise along the length of the heat exchanger were then used, in conjunction with the measured mass flow and the temperature difference between the two streams of liquid metal, to determine the overall heat-transfer coefficient.

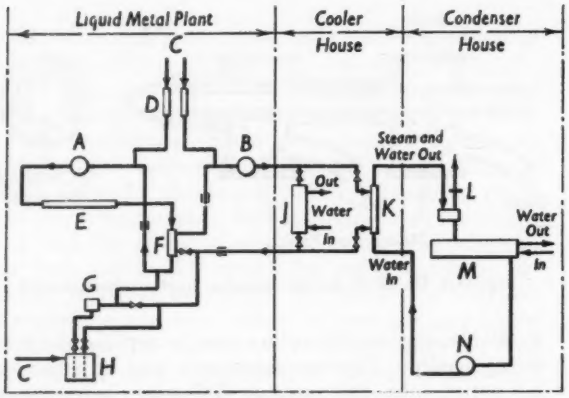
The method of filling the apparatus, the level control of the free surfaces, and the oxide removal equipment were basically the same as for the pump circuit described above.

Apparatus for Experiments at High Heat Fluxes

The main purpose of this experiment was to investigate the effect of high heat fluxes, and consequently high temperature gradients and thermal stresses, on tubes of various materials. A secondary purpose to which the apparatus has been put is the provision of testing facilities for various small liquid-metal-to-liquid-metal and liquid-metal-to-water heat exchangers. The experimental results obtained with this apparatus are not within the scope of the present paper, but the design and operation of the equipment may be of interest.

The apparatus is shown in diagrammatic form in Fig. 4. Firstly, it will be seen that the equipment for storing the liquid metal and supplying it to the apparatus is basically similar to that in the two circuits which have already been described. Nitrogen is used to displace the liquid metal from the reservoir tank into the apparatus by way of a sintered stainless steel filter; the nitrogen with which the apparatus was initially filled is simultaneously released from the level-control vessels at the top of the system.

The experimental apparatus consists of two separate 1 in. diameter pipe circuits, each with its own single phase electromagnetic pump linked together by a small tube-and-annulus heat exchanger. The first circuit, containing pump *A*, is provided with an electric heater of about 100kW capacity; the second,



—▷◁— Valves shown on liquid-metal circuits only.
—A— Single-phase electromagnetic pump.
Electromagnetic flow meters.
Direction of flow.

A, B—Pumps.
C—Nitrogen supply.
D—Level control vessels.
E—Heater.
F—High flux exchanger (see
G—Filter.
H—Reservoir tank.
J—Cooler.
K—Evaporator.
L—Safety valve.
M—Condenser.
N—Recirculating pump.

Figure 4. Layout of high heat flux apparatus.

containing pump *B*, includes a cooler in which heat is rejected to water. A sketch of the tube-and-annulus heat exchanger is shown in Fig. 5; the hot liquid metal passes through the tube and the cooler liquid metal through the annulus. It has been possible to produce a heat flux through the wall of the tube of almost 1kW per square centimeter (approximately 3×10^6 B.T.U. per square foot per hour). If this tube is of stainless steel, 0.020 in. thick, the heat flux corresponds to a conduction temperature drop in the thickness of the tube of about 250 deg. Cent.; the consequent thermal stresses, based on elastic theory, are of the order of 40 tons per square inch.

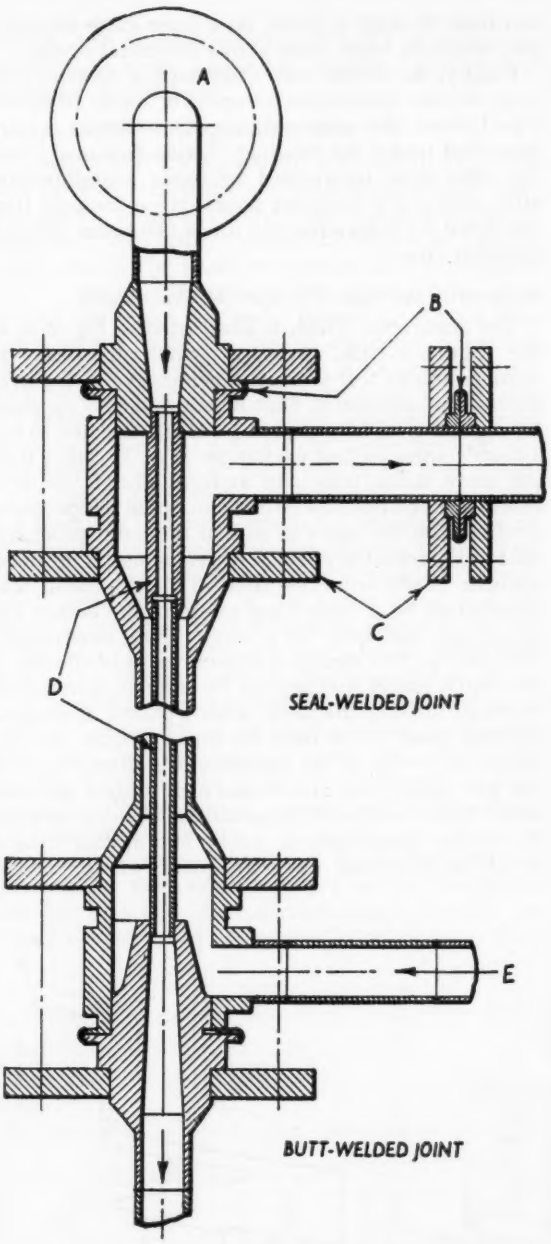
The water cooler is situated in an annex to the main building so as to limit the amount of liquid metal and water which can come into contact in the event of an accidental leakage. The heat exchanger itself consists of separate banks of liquid-metal and water tubes bonded together by casting in a block of aluminum. An alternative cooler, consisting of a simple tube and annulus heat exchanger with boiling water in the tube, is also available.

DESIGN AND OPERATIONAL PROBLEMS

The design of the various components which are common to most liquid metal circuits will now be considered in more detail.

Storage

The first consideration in the handling of sodium and sodium-potassium alloys is the problem of storage. The alloys containing between 15 and 50 per cent



Showing the methods of pipe jointing.
A—Hot inlet.
B—Seal welds.
C—Mild steel backing flanges and bolts.
D—Distance piece to adjust length of test tube.
E—Cold inlet.

Figure 5. Tube-and-annulus heat exchanger for high heat fluxes.

sodium by weight are liquid above 10 deg. Cent. Outside this range there will at this temperature be a mixture of solid sodium or potassium and the eutectic (which contains 22 per cent sodium and freezes at —12 deg. Cent.). The supply of material for the present work consisted of solid sodium in the form of 3½

lb. bricks packed in mild steel drums with airtight lids, and the liquid eutectic in mild steel carboys with screwed caps, each containing 112 lb. Alloys containing more sodium have been prepared and stored in the same carboys; in this case it is necessary to heat the carboys above the temperature on the liquidus curve corresponding to the particular alloy before discharging.

The storage area should be dry and there should be no water supply in the building. A supply of dry soda ash or one of the proprietary brands of fire-extinguishing powder for alkali metals should be readily available. It sometimes happens that if metal is exposed to a humid atmosphere before sealing in its container, a pressure of hydrogen is developed; there is, therefore, the possibility of hydrogen explosion if the container is opened near a naked flame. It is usual to provide shallow steel trays to contain accidental spillages, which are then dealt with by covering with soda ash and removing to a disposal area where they are burnt. Neither sodium nor sodium-potassium alloys will burn freely even in moist air unless heated above 100 deg. Cent.

Materials of Construction

There is a considerable amount of information in the literature concerning the compatibility of various materials with sodium and sodium-potassium. (Sodium and the alloy are usually considered to be similar in this respect.) The choice of materials is very wide provided that a temperature of 200 deg. Cent. is not exceeded (*Liquid Metals Handbook,* 1952). Mild steel is suitable for long-term use up to 450 deg. Cent., and for higher temperatures suitable materials include the austenitic and ferritic stainless steels, pure iron, nickel, inconel, nichrome, and the refractory metals niobium, molybdenum, tantalum and tungsten.

In the work described in this paper, an austenitic 18/8 titanium stabilized stainless steel was used. Although mild steel has been used for lower temperature work, stainless steel is in general preferable for experimental apparatus where it is necessary to open the circuit to atmosphere frequently without oxidation of the pipe surfaces.

Fabrication Methods

An all-welded construction is recommended for long-term operation, since the choice of jointing materials to withstand alkali metals is strictly limited, and normal brazing materials are unsuitable. For short-term operation at temperatures up to 400 deg. Cent. a flanged joint with a soft copper joint ring may be used and for temperatures up to about 150 deg. Cent. various proprietary types of metal-to-metal coupling have been found sastisfactory. The liquid metal possesses the doubtful virtue of sealing itself by oxidation against small leaks at low pressure and temperature.

Extensive use has been made of the type of seal-welded flanged joint which is shown in Fig. 5. It is not recommended for general use since the number of times the weld can be removed and the joint re-sealed is limited to about three or four. The thermal capacity of the rather bulky flange is also an embarrassment when the pipe system has to be heated by electrical trace heaters before filling with sodium.

A butt-welded pipe joint is recommended whenever possible for the following reasons. First, it may be radiographed, and, secondly, the circuit may be opened by a saw cut and sealed again by a butt weld. Argon arc fusion welds are used, the pipe being filled with nitrogen before welding so as to prevent oxidation at the back of the weld. For larger welds a filler rod of the same material as the pipe is used, or, alternatively, the initial argon-arc weld is backed up by metallic-arc runs. Heat-treatment of the weld has been found unnecessary.

In the earlier stages of the work all welds were radiographed, but this condition is now relaxed on equipment for short-term operation. In the course of the whole of the work described in this paper, there has not been a single failure of a butt-welded joint.

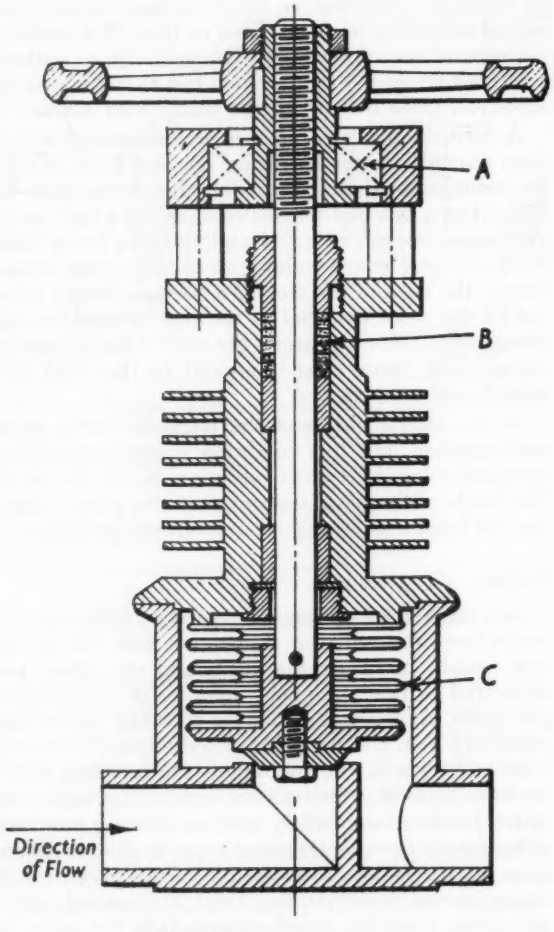
Valves

One method of solving the valve problem has already been described in connection with the pump test circuit, in which the valve and the gland are separated so that the gland operates in the blanket gas space. Another expedient is to make use of the relatively high melting point of sodium and to freeze a short length of the sodium in the pipeline. Both methods have their limitations; the former leads to a rather cumbersome design, and the packing may still suffer attack from liquid metal vapor in the gas space, although this is not serious with asbestos packing and temperatures below 150 deg. Cent. The second method suffers from the disadvantages that the valve is either fully open or fully closed, and that the time required to thaw the metal may be an embarrassment under certain operating conditions.

A design of valve in which the gland is protected from the liquid metal by flexible metal bellows is shown in Fig. 6. In the case of sodium, an additional measure of protection is afforded by the narrow annulus, cooled by external fins, in which the sodium will freeze before reaching the gland; this does not apply for alloys which are molten at room temperature. A valve of this type has given trouble-free service with sodium-potassium alloy at 250 deg. Cent.

Pumps

Electromagnetic pumps have been used throughout the work reported here, except for the heat-transfer apparatus in its original form. They have a considerable advantage over mechanical pumps in that there are no moving parts and no glands. It should, however, be mentioned that they tend to become inefficient when used on liquid metals of low conductivity, such as lead-bismuth alloys. The gland problem of the mechanical pump may be avoided by



A—Bearing.
B—Graphited asbestos packing.
C—Bellows.

Figure 6. 1 in bellows valve.

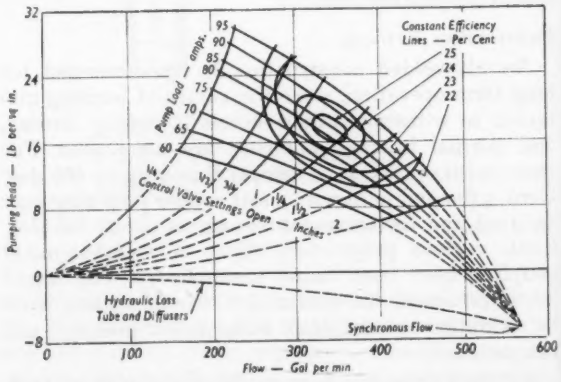
using the "canned rotor" principle, in which the armature of the motor runs in liquid metal and is separated from the field windings by a thin metal sheath. Such a pump, of 50 h.p. capacity, has been designed but has not yet been tested (Fortescue, 1954). It may be possible, with a carefully designed mechanical pump to attain an efficiency of 60 to 70 per cent, whereas an electromagnetic pump operating under favorable conditions will probably not exceed 40 to 50 per cent. Nevertheless, in most applications the lower efficiency of the electromagnetic pump will be amply compensated for by its greater simplicity.

The various types of electromagnetic pump have been described by Woollen (1955), and a 400 gallons-per-minute flat linear induction pump (F.L.I.P.) was tested in the 4 in. diameter circuit described above. The F.L.I.P. operates on the same principle as an induction motor; the windings are arranged so that they produce a traveling magnetic field along the length of a narrow rectangular channel, and the liquid metal in the channel corresponds to the armature of the induction motor. An alternative type of pump, which is used on the high-heat-flux circuits described above, is the conduction pump; in this design a current is passed through the liquid metal in a direction perpendicular to its direction of flow, and a magnetic field is provided in the third mutually perpendicular direction. This type of pump is normally used only in small sizes, and its efficiency is usually in the range 5 to 10 per cent. For a more detailed description of these and other types of electromagnetic pump reference should be made to Woollen (1955).

Fig. 7 shows the experimentally determined characteristic curves of the 400-gallons-per-minute F.L.I.P. Lines of constant efficiency have been superimposed on the pressure rise against flow curves for various winding currents. One aspect of the operation of this type of pump is worthy of mention. There is normally a contraction of area from the pipeline to the narrow rectangular duct of the pump and, consequently, the pressure in the duct will be less than that in the pipeline by an amount equal to the losses in the section changer and the difference in kinetic pressure at the two points. If the duct pressure becomes sub-atmospheric, the rather flexible walls deflect inwards, thus increasing the velocity and reducing the pressure still further (provided the pipeline pressure remains constant). In practice, when a pump is operated in this region, there is a drastic reduction in flow and pressure rise.

Level Measurements

Devices for measuring the level of the liquid metal are required in reservoir tanks, expansion tanks, and in manometer tubes. In many cases it is necessary to know only whether the liquid surface lies between two fixed points; for example, in an expansion vessel at the top of an apparatus. The level in a manometer, on the other hand, must be measured accurately; in some cases it is also desirable to measure the level in a reservoir tank accurately, since it is then possible to follow the process of filling the apparatus more closely.



Flat linear induction pump, sodium-potassium eutectic at 250 deg. Cent.

Figure 7. Pump characteristics.

Probably the simplest form of fixed-level indicator is the insulated probe, which is inserted through the top of an expansion chamber or other vessel; the probe is insulated from the tank lid, and when the liquid metal makes contact with it an electrical circuit is closed which lights a warning lamp on the control panel. Several probes of different lengths may be used to determine the level within closer limits. This method has been widely used on the apparatus described above; a normal sparking plug is used to provide the gastight, insulated connection through the tank lid. In the main this type of indicator has proved satisfactory although at higher temperatures some trouble has occurred as a result of condensation of metal vapor on the insulator, with subsequent "shorting" of the probe. An alternative which does not involve direct connection with the liquid metal, is to employ a gamma-ray absorption method. A source of gamma-rays is placed at one side of the vessel and a detector at the other; the difference in the attenuation of radiation from the source when liquid metal is interposed between it and the detector can be made to operate an indicator lamp or warning bell. The sensitivity of this method is limited by the relative attenuation of the radiation in the walls of the vessel and in the liquid metal; it has been used quite satisfactorily on a 3 in. diameter stainless steel vessel with walls 0.128 in. thick. It is ideally suited to level measurements at high temperatures.

The level of a liquid metal in a manometer tube may be measured by one of the following two methods: in the first, which was used on the heat-transfer apparatus described above, a direct current of about 1A was passed down the tube and measurements of the potential along its length were used to determine which part of the tube was "shorted" by the liquid metal inside it. Voltage tappings were connected to the tube at 10 cm. intervals along its length, and adjacent tappings could be connected across a galvanometer by means of a selector switch; the potential difference across pairs of tappings below the liquid metal surface was less than that for an empty tube, hence the two tappings between which the surface lay was quickly determined. An estimate of the exact position of the surface between the tappings was then made by comparing the potential difference between them with the corresponding values for an empty tube and for a full tube.

The second method depends upon the fact that the inductance of a coil of wire placed round the pipe is affected by the presence of liquid metal in the pipe. If the coil is moved along the pipe it is possible by this means to detect the position of the surface; the manometer tubes across the nozzle flow meter in the pump testing circuits were fitted with this type of instrument. It should be noted that since the method depends upon the eddy currents generated in the liquid metal (which in the case of sodium is non-magnetic), the tube should be of non-magnetic material or it will shield the liquid metal inside it.

The level in the reservoir tank of the pump testing circuit was measured by means of a float which was magnetically coupled to an indicator dial. The float contained a permanent magnet and was free to slide up and down a vertical tube passing through the tank lid. The bottom of this tube was sealed and a magnet was suspended inside the tube by a wire which operated a balanced indicator dial. Movement of the float and its magnet thus operated the indicator by way of the magnet inside the pipe. Oxide contamination of the liquid metal was responsible for most of the troubles experienced with this instrument; unfortunately, the reservoir tank is often the coldest part of the circuit, and oxide tends to separate out there. This emphasizes the importance of the equipment for the continuous removal of oxide, which is described later.

Flow Measurement

The basic method of flow measurement in all the equipment described here is by means of either a nozzle or Venturi meter designed in accordance with B.S. 1042. Electromagnetic flow meters are calibrated against these either in the same apparatus or in a special calibration circuit.

The electromagnetic flow meter consists of a section of pipe across which a magnetic flux is maintained, usually by means of a permanent magnet. Potential tappings are connected to the pipe so as to measure the potential across a diameter of the pipe at right angles to the magnetic flux. The flow through the pipe is proportioned to the e.m.f. developed at the tappings and the constant of proportionality for a given flow meter is determined by comparison with a nozzle or Venturi meter. Elrod (1952) has developed a theoretical expression relating the e.m.f. from the flow meter with the flow, the strength of the magnetic flux, the dimensions of the system, and the resistivities of the liquid metal and the pipe material. Tests in connection with the apparatus described in this paper have shown that, while the results confirm the general form of his expression, the magnitude of the e.m.f. obtained in practice is about 80 per cent of the value predicted.

Cottrell (1953) has described the use of rotameters and rotary-vane flow meters with liquid metals.

Pressure Measurement

Conventional Bourdon tube pressure gauges (with stainless steel tubes) have been used where it is possible to ensure that the liquid metal is not in direct contact with the tube. This may be achieved by interposing a vessel between the liquid metal and the gauge in which an inert gas is trapped; such an arrangement is shown in Fig. 1. Diaphragm gauges are preferable to Bourdon tube gauges and may be used in direct contact with the liquid metal, although this is not recommended, since it is then necessary to keep the diaphragm at a sufficiently high temperature to prevent oxide deposition. Where it is inconvenient to situate the pressure gauge close to the point where the pressure is to be measured, the gauge

may be connected to this point by means of a capillary tube filled with an inert liquid, which is then sealed from the liquid metal by a diaphragm. The diaphragm assembly may be heated to the temperature of the liquid metal circuit to prevent oxide deposition. The intermediate liquid may be silicone oil, for low-temperature operation, or sodium-potassium eutectic for high temperatures. This type of instrument has given satisfactory service on the high heat flux apparatus described above. Cottrell (1953) described the use of flexible metal bellows in place of the diaphragm.

Oxide Removal

Oxidation of sodium or sodium-potassium alloy in an apparatus may be caused either by oxygen in the gas used to blanket the free surfaces, or by reduction of metallic oxides on the surfaces of the pipes. While it is possible to reduce contamination from both of these causes it is preferable, in addition, to provide for the continuous removal of oxide from the liquid metal.

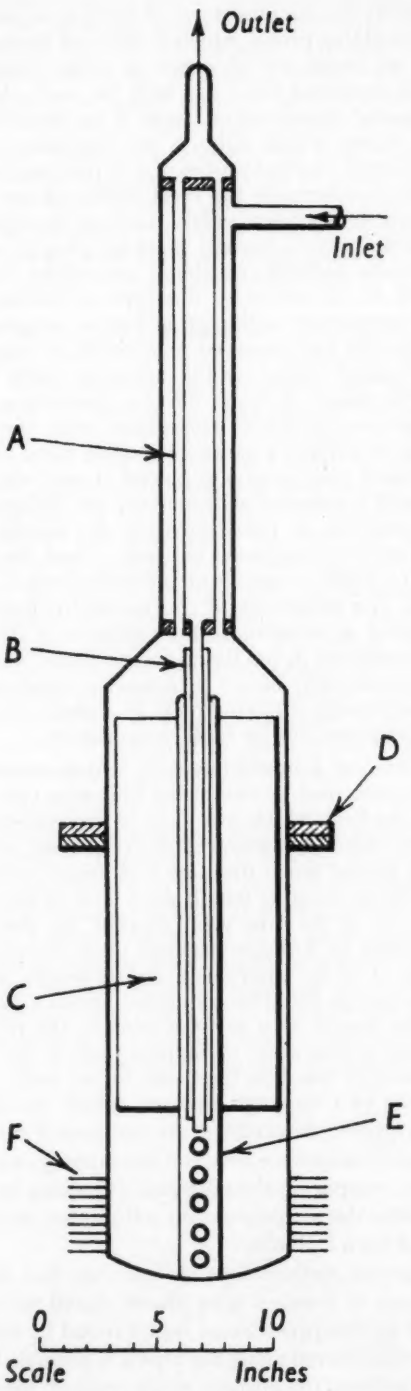
The effects of oxide contamination are twofold: firstly, the presence of oxide materially affects the corrosion problem, and secondly, the contamination may be sufficiently severe to cause blockage of the pipes.

Horsley (1954) has shown that sodium oxide contributes largely to the corrosion of steel by sodium and sodium-potassium alloys as a result of the formation of complex iron sodium oxides.

The solubility of sodium oxide in both sodium and sodium-potassium is small but increases with temperature, so that solution of oxide occurs in high-temperature regions with deposition in low-temperature regions. Thus if no steps are taken to remove it, oxide may build up locally at the coolest part of the circuit and cause a blockage. Fortunately, this transfer process provides a convenient method for the continuous removal of oxide; a small fraction (approximately 1–2 per cent) of the liquid metal is passed through a vessel which is maintained at a lower temperature than the rest of the circuit, and the oxide is deposited in this vessel. A typical cold trap, which was installed on the pump testing circuit (Fig. 1) is shown in Fig. 8. The trap operates at a temperature of about 150 deg. Cent., with the main circuit at about 250 deg. Cent. A window was fitted to this circuit above the free surface in the control valve. Before the cold trap was fitted the surface was generally covered by a layer of oxide, and in fact it was necessary to clean out the apparatus after 360 hours of operation; after fitting the cold trap the free surface remained clean and a run of 10,000 hours' duration was achieved with no sign of contamination.

HEAT-TRANSFER MEASUREMENTS

The apparatus used for this work has been described above. At the time when the work was initiated there was available a considerable amount of heat-transfer data for mercury, some for lead-bismuth alloys, and a small amount for sodium-potassium alloy. The extremely large discrepancies be-



See Fig. 1 for location.

A—Regenerative heat exchanger containing six equally pitched tubes.
B—Double-wall center tube.
C—Basket carrying lessing rings with perforated bottom plate welded to perforated tube.
D—Flange with annealed-copper joint-ring.
E—Perforated tube.
F—Copper fins brazed to stainless steel shell.

Figure 8. Cold trap.

tween the results of various workers using mercury had not been resolved, and there was considerable controversy as to whether wetting of the solid surface by the liquid metal had any effect on the heat transfer. For example, Trefethen (1951) carried out experiments with mercury under wetting and non-wetting conditions and found no difference between them; on the other hand, Doody and Younger (1951), again using mercury, found such an effect. More recently it has been stated that a reduction in heat-transfer coefficient under non-wetting conditions may be suppressed by an increase in the pressure of the liquid metal, suggesting that some type of cavitation phenomenon is responsible for loss of contact between the surface and the liquid metal (Stormquist, 1953).

In these circumstances, it was thought desirable to obtain basic heat-transfer measurements under conditions which would be dictated by experience in operating sodium circuits; for example, it appeared unlikely that the high standard of cleanliness which is possible in laboratory apparatus could be maintained when dealing with plant-scale equipment.

The results obtained on the heat-transfer apparatus are shown in Fig. 9, where they are compared with a curve predicted by Lyon (1951). A corresponding equation for circular tubes proposed by Lyon, and based on theoretical work by Martinelli (1947) is

$$Nu_t = 7 + 0.025\,(Re \,.\, Pr)^{0.8} \quad . \quad . \quad . \quad . \quad (1)$$

and for an annulus

$$Nu_a = 0.75 Nu_t \left(\frac{r_o}{r_i}\right)^{0.3} \quad . \quad . \quad . \quad . \quad (2)$$

Equation (2) is represented by the curve in Fig. 9 and is confirmed by both the sodium and sodium-potassium alloy results for Reynolds numbers in the turbulent-flow region. It should be noted that equation (1) is likely to be inadequate at lower Reynolds numbers, since liquid metals of different Prandtl numbers could have Reynolds numbers in the laminar and turbulent Reynolds numbers regions respectively, and yet have the same value of Peclet number ($Re \,.\, Pr$).

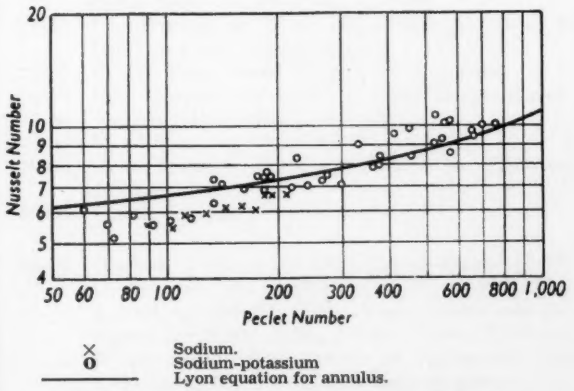
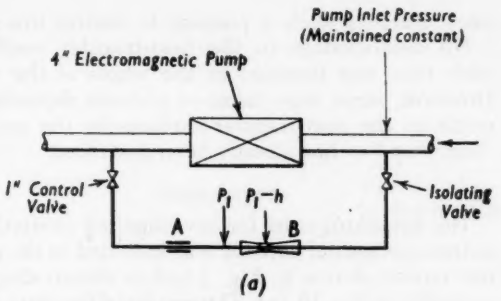
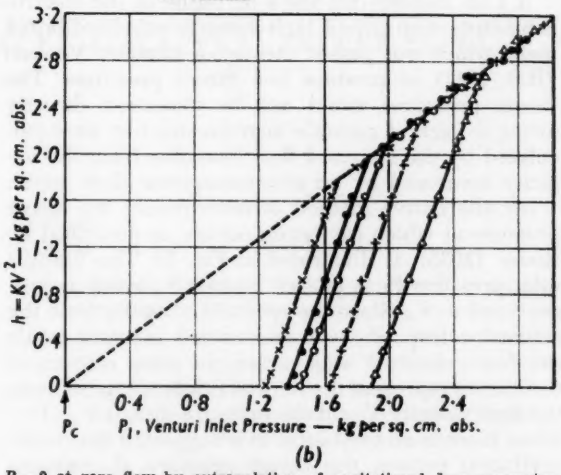


Figure 9. Results from double-annulus exchanger.



(*a*) Diagram of test circuit.
A—Electromagnetic flow meter.
B—Venturi.



P_c—0 at zero flow by extrapolation of cavitational flow parameter.
P_i—Pump inlet pressure at zero flow.
Pump inlet pressure is maintained constant.
– – – – – – Cavitational flow.
——— Normal flow.
$P_c = P_i - h$.
× P_i=1.23kg per square centimeter.
● P_i=1.37kg per square centimeter.
O P_i=1.43kg per square centimeter.
\+ P_i=1.63kg per square centimeter.
Δ P_i=1.89kg per square centimeter.
(*b*) Curve showing relation between h and P_i.
h—KV^2.
K—Venturi constant.
V—Velocity of flow.

Figure 10. Cavitation tests with sodium-potassium eutectic at 250 deg. Cent.

In the course of the work it was found that a slight eccentricity of the annuli produced a much larger temperature variation in a circumferential direction than would be expected with fluids of higher Prandtl number (that is, gases and non-metallic liquids). This effect is discussed elsewhere (Hall and Jenkins, 1955). It should be borne in mind when assessing the thermal stresses likely to be set up in a liquid-metal heat exchanger.

A few measurements were made at Reynolds numbers in the laminar-flow region, and instead of giving the theoretically predicted constant Nusselt number the results showed a continued decrease as the Reynolds number decreased. The classical theory, upon which the prediction of constant Nusselt number is based, neglects conduction in the direction of flow, and it is suggested that this may explain the discrep-

ancy. Further work is planned to resolve this point.

No deterioration in the heat-transfer coefficient with time was detected in the whole of the work. However, steps were taken to prevent deposition of oxide on the heat-transfer surfaces by the use of a "cold trap," as has already been described.

CAVITATION

The apparatus used for investigating cavitation in sodium-potassium eutectic was included in the pump test circuit shown in Fig. 1 and is shown diagrammatically in Fig. 10 (*a*). The results of the tests at 250 deg. Cent. are shown in Fig. 10 (*b*).

A 1 in. tapping into the 4 in. outlet of the electromagnetic pump gave a high-velocity supply of liquid metal which was passed through a standard Venturi (B.S. 1042) to produce low throat pressures. The throat pressures could not be measured directly owing to lack of suitable instruments, but were calculated by the standard flow formulae from the velocity measured by an electromagnetic flow meter.

An alternative method of determining the throat pressure at which cavitation occurs, as described by Rouse (1938), is illustrated in Fig. 10. The Venturi inlet pressure P_1 is plotted against h, which is proportional to V_2. Since the pressure at outlet from the cavitation loop, P_i, was maintained constant while the flow velocity V was varied, the steep sections of the curves represent the normal relationship between the flow velocity V and the pressure drop (P_1—P_i), when there is no cavitation. It is suggested that when cavitation occurs, the throat pressure P_c remains constant at the vapor pressure of the liquid, that is,

$$P_1 - KV^2 = P_i = \text{constant} \quad . \quad . \quad . \quad . \quad (3)$$

Under these conditions the relationship between P_1 and K is unaffected by changes in the downstream pressure P_i; moreover, if this section of the graph is extrapolated to zero flow, the intercept on the P_1 axis will give the cavitation pressure P_c. (If $V=0$ in equation (3), then $P_c=P_1$). It will be seen that the results obtained at different values of the downstream pressure P_i, do conform to this pattern. The virtue of this method lies in the fact that the absolute accuracy of measurement of the flow velocity V is not important in determining the cavitation pressure P_c. It will be seen that this method leads to a value of P_c very close to zero; the accuracy of the measurements is not such as to distinguish between zero pressure and the vapor pressure of the liquid metal (the vapor pressure of sodium-potassium eutectic is 1 mm. of mercury at 355 deg. Cent.).

REFERENCES

Cottrell, 1953, Technical Information Service, Oak Ridge, Tennessee, ORNL 1688, "Sodium Plumbing."

Doody and Younger, 1951, United States Atomic Energy Commission, NP 1788, "Heat Transfer Properties of Liquid Metals."

Elrod, 1952, *Trans. A.S.M.E.*, Vol. 74, No. 4.

Fortescue, 1954, *Journal Nuclear Energy*, Vol. I, No. 1.

Hall and Jenkins, 1955, *Journal Nuclear Energy*, Vol. I, No. 4.

Horsley, 1954, A.E.R.E. M/R 1441.

Liquid Metals Handbook, second edition, NAVEXOS, P-733 Rev. (U.S. Government Printing Office).

Martinelli, 1947, *Trans. A.S.M.E.*, Vol. 69, Page 947.

Rouse, 1938, *Fluid Mechanics for Hydraulic Engineers*, page 132 (McGraw-Hill, London and New York).

Stromquist, 1953, Technical Information Service, United States Atomic Energy Commission, O.R.O. 93, "Effect of Wetting on Heat-Transfer Characteristics of Liquid Metals."

Trefethen, 1951, United States Atomic Energy Commission, NP 1788, "Heat Transfer Properties of Liquid Metals."

Woollen, 1955, *Fluid Handling*, Nos. 62 and 63.

J. W. SAWYER

ENGINEERING ABSTRACTS

Abstract Number	*Abstract Title*
58–56	Special Turbine for High-Speed Running
59–56	Studies in Corrosion
60–56	Effect of Nuclear Radiation on Structural Materials
61–56	Packing and Gasket Materials
62–56	Stress Relief Heat Treatment Equipment for Welds of All Kinds
63–56	Fundamentals of Electrical Discharge Machining
64–56	Arc Supression with Semiconductor Devices
65–56	The Planning of Systems and Products for the Military Services
66–56	Thermonuclear Power — Man's Most Glittering Dream
67–56	High Speed Photography
68–56	GMR 4-4 Hyprex Free Piston Turbine Engine
69–56	A Review of Developments in Plastics Engineering—1954-1955
70–56	Fire-Resistant Lubricants and Hydraulic Fluids
71–56	Characteristics of Nuclear Reactors — A Staff Report
72–56	Welding — An Old Art Meets New Needs in Aircraft Fabrication Today
73–56	A Review of Some Combustion Problems Associated with The Aero Gas Turbine
74–56	Prestressing an Ultra High-Strength Steel to Perform Even Higher Duty
75–56	British Free-Piston Engine Development
76–56	How Graphite Performs at High Temperatures
77–56	Applying Radiation in Industry
78–56	Metal Spraying with Rockets
79–56	Heat Transfer Problems Solved by Electrical Analogy
80–56	Plastic Springs

58–56 "Special Turbine for High-Speed Running"—R. C. McLeod; The Engineer, V. 201, n. 5232, May 4, 1956, pp 434-436. This steam turbine was designed for 12,500 r.p.m., 350 psi steam, 15,000 h.p. It was required to operate at any speed from 4,000 to 12,500 r.p.m. and deliver power from 1,000 to 15,000 h.p. The speed is controlled by hand. Due to the low weight of the rotor with respect to the power the rotor can be brought from zero to 12,500 r.p.m. within two seconds. Blade failures due to vibration have been eliminated by using higher strength material and lacing wire at the tips. The loss in performance due to installation of the lacing is not known. However, no drop in performance has been noted.

59–56 "Studies in Corrosion"—G. H. Cartledge; Scientific American, V. 194, n. 5, May, 1956 pp 35–39. A new element, technetium, is proving to be an outstanding rust inhibitor. It is a synthetic element found in the products of uranium fission and is one of the rarest of all elements on earth. Some of the unanswered questions on corrosion may be solved as a result of tests and studies being conducted with technetium. The mechanism of corrosion is discussed and illustrated. This element may never be plentiful enough for general use as a rust inhibitor. Its supply may be adequate for use in certain areas in nuclear reactors. As a research tool it offers promise for work involving the properties of metals.

60–56 "Effect of Nuclear Radiation on Structural Materials"—J. C. Wilson and D. S. Billington; Journal of Metals, V. 8, n. 5, May, 1956, pp 665-672. In the selection of materials for nuclear reactor design the effect of nuclear radiation on the engineering properties of a material should be considered. Criteria peculiar to nuclear reactors are presented. Physical properties, strength, hardness, ductility, toughness and creep are discussed. Radiation effects on mechanical properties of various alloys and metals are reviewed. Tables, curves and references are given.

Abstract Number — *Abstract Title*

61–56 "Packing and Gasket Materials" — Robert H. Koch; Machine Design, V. 28, n. 11, May 31, 1956, pp 117–120. Some of the recently developed packing and gasket materials that show promise, yet are not widely used, are discussed. Plastics and elastomers offer interesting materials. In the plastic materials polytetrafluoroethylene (PTFE) and monochlorotrifluorethylene (MCTFE) are reviewed. Plastic materials include: PTFE felt impregnated with PTFE dispersion; fabric laminates impregnated with fluorocarbon dispersion; filled fluorocarbon plastic; teflon fiber packing impregnated with teflon dispersion; kel-F coated silicone-rubber o-rings; machined teflon envelope gaskets; bonded-teflon fabricated gaskets. The elastomers are: hypalon chlorosulfonated polyethylene; kel-F fluorocarbon elastomer; polyacrylic rubber; silicone rubber; urethane rubber. These new materials hold promise for many applications where corrosive liquids, high or low temperatures, oils, acids, oxidizers, wear and other problems are encountered.

62–56 "Stress Relief Heat Treatment Equipment for Welds of All Kinds"—K. Frauenfelder; The Brown Boveri Review, V. 43, n. ½, Jan/Feb 1956, pp 48–57. The stress relief of heavy duty weldments is essential. Induction heating methods have been developed that permit stress relieving of intricate shapes during field work without use of annealing furnaces. Methods of stress relieving numerous parts are described and illustrated. These include: built up steam turbine shafting; penstock; steam pipe; valve box on a high pressure steam turbine; flange on a water evaporator; intake pipe on a Francis turbine; a Pelton wheel; shrinking a gear wheel on a shaft with a double pinion. Construction of the heating equipment is described and illustrated.

63–56 "Fundamentals of Electrical Discharge Machining"—Machinery, Part I, V. 62, n. 9, May 1956, pp 139–145; Part II, V. 62, n. 10, June 1956, pp 169–176. The electrical discharge machining method is widely used in Russia. This series of articles is based on translations of Russian technical papers and operations. The following are discussed: electrical circuits necessary for the process, electrode materials, working fluids, productivity, surface finish, tool materials and design, methods of manufacturing various electrical discharge machining tools, drilling with electrical discharge method and various operations.

64–56 "Arc Supression with Semiconductor Devices"—F. W. Parish; Electrical Manufacturing, V. 57, n. 6, June 1956, pp 127–131, 344, 346. The problem of protecting contacts that interrupt current flow is important. The protection of contacts can be accomplished by blow-out coils, permanent magnets, fixed resistors, capacitors and semiconductor devices. Design procedures for using germanium or selenium rectifiers to provide contact protection are presented. Curves, photographs, tables and references are included.

65–56 "The Planning of Systems and Products for the Military Services"—L. L. Stahl; Aeronautical Engineering Review, V. 15, n. 6, June, 1956, pp 63–65. Military needs, systems and products, and their related problems are discussed from the point of view of business and management. The planning is considered in two phases: market intelligence and analysis of the planning process. Market intelligence determines: location; need; relationship to current and future use; identity of the military planners; propriety of location; developing a true need; size of market; service attitudes and preference. Analysis of the planning process: determine steps in the planning; examine the company with respect to the product or system under consideration.

66–56 "Thermonuclear Power — Man's Most Glittering Dream" — Andrew W. Kramer; Power Engineering, V. 60, n. 6, June 1956, pp 82–85, 108, 109. The possibility of development of controlled thermonuclear reaction offers tremendous advantages. It may have far greater importance than worldwide use of atomic energy from fission. Some estimates have been made that a controlled fusion process is not closer than 25 years. The Mckinney Report states "it is virtually certain that no full-scale (thermonuclear) reactor will be developed in the next year or two; it is, however, probable that success will be achieved eventually." Stellar thermonuclear reactions and controlled reactions are discussed.

67–56 "High Speed Photography"—W. G. Hyzer; The Tool Engineer, V. XXXVI, n. 6, June, 1956, pp 67–72. Modern high speed motion picture cameras and techniques can be used effectively in analysis of production operations. The analytical capabilities of the eye can be extended up to 500 times with the high speed camera. Some cameras are capable of taking 16 mm pictures at the rate of 8000 frames per second. Minimum practical projector frequency is approximately 15 frames per second. Photographic techniques and computations are presented. Practical usage includes: study of tool design and machining operation, including cutting action, chip flow and coolant flow; tool cutting characteristics in turning operation; punch press operation; and arc welding. Tables, photographs, drawings and formulae are included.

68–56 "GMR 4-4 Hyprex Free Piston Turbine Engine" —A. F. Underwood; SAE Journal, V. 64, n. 7, June 1956, pp 60–66. A 250 hp free piston-gas

Abstract Number *Abstract Title*

turbine engine has been designed, built and installed in a passenger car. Results indicate it will be a serious contender for powering automobiles. Data on the free piston gasifier is as follows:

Horsepower (gas)	250
Length in.	40
Width in.	34
Height in.	18
Cycle speed	
Max. strokes/min	2400
Min. strokes/min	1000
Specific fuel consumption lb/gas hp-hr	0.45

The free piston gasifier, which is a siamesed twin cylinder unit, is located under the hood. The gas turbine and transmission are in the rear. Advantages of this power plant include the following: various fuels can be used: kerosene, No. 2 diesel oil, Bunker C, whale oil, peanut oil; overall efficiency is high: 32–36 percent including the turbine; high torque available from the turbine at low speeds; gas entering the turbine is relatively cool: 450°-900°F; response to throttle is rapid; additional power augmentation can be obtained with afterburner; suitable for all types of vehicles.

69–56 "A Review of Developments in Plastics Engineering—1954-1955"—Bruce Maxwell: Mechanical Engineering, V. 78, n. 6, June 1956, pp 533–535, 540. Domestic production of plastic was 2776 million pounds, 35 cents per pound, in 1953 and 2827 million pounds in 1954, 34 cents per pound. The annual rate for the first six months of 1955 was 3020 million pounds. New materials and new formulations of old materials include: epoxy resins, silicones, styrene copolymers, ethylene glycol film and tape, nylon, isocyanate resins, ethylenepolymers, and low-pressure polyethlene. Improvements were made in processes of fabrication such as extruder performance, vacuum-molding and use of phenolic and epoxy molds. Products that are new: glass-reinforced moldable sheet: plastic tools for metal forming; plastic pipe and plastic lining for metal pipe, decorative plastics; tiny microballoons of phenolic floating on the surface of oil in storage tanks that eliminate 80–90 percent of evaporation.

70–56 "Fire-Resistant Lubricants and Hydraulic Fluids"—R. B. McBride; General Electric Review, May - July 1956, V. 59, n. 3–4, pp 51–53. Flammable fluids used in hydraulic equipment have been ignited after failures of high pressure elements on aircraft carriers and aircraft. Accidents of this type also occur in die-casting, glass drawing machines, presses and automatic welders. Steam turbine grade petroleum oils have spontaneous-ignition temperatures in the range 700-800 F. Failure of a turbine lubricating oil line could result in serious fire damage. Fire prevention devices designed into a turbine may cost $40,000 and still not eliminate fire hazard. Development of several fire resistant fluids for use in turbines and hydraulic equipment holds promise. Two power companies are now testing modified phosphate-ester fluids in boiler water feed pump turbines and one fluid is being tested in a large turbine generator unit.

71–56 "Characteristics of Nuclear Reactors — A Staff Report" — Product Engineering, V. 27, n. 5, May 1956, pp 138–147. Nuclear reactors are discussed and classified according to the following points: homogeneous or hetrogeneous; type of fuel; energy range: fast, thermal or intermediate; use, power, production reactor, research; type of moderator: graphite, beryllium oxide, beryllium carbide, light water, heavy water; type of shielding: water, concrete, steel, lead; type of coolant: liquid metal, light water, heavy water, gas; type of control; reactivity control rod, boron, cadmium, hafnium; heat and electrical output. References, diagrams, and illustrations.

72–56 "Welding — An Old Art Meets New Needs in Aircraft Fabrication Today"—Alvin L. Bennett; Solar Blast, V. 17, n. 2, June, 1956, pp 8–14. Some of the basic types of welding used in aircraft fabrication are reviewed. They include: oxyacetylene, arc, metallic arc, submerged arc, inert gas shielded arc, sigma, atomic hydrogen, carbon arc, forge and resistance welding. Advantages, disadvantages and equipment used in these various types are discussed. The important part played by the operator is emphasized. Training of operators and careful quality control are essential to assure satisfactory welds.

73–56 "A Review of Some Combustion Problems Associated with The Aero Gas Turbine" — J. S. Clarke; The Journal of The Royal Aeronautical Society, V. 60, n. 544, April 1956, pp 221–240. This paper is based on experiences and studies on the burning of atomized sprays. Subject material presented includes: operating conditions of combustion chambers; flame stabilization and heat release; comparison of pipe, tube and annular chambers and upstream and downstream fuel injection; combustion problems, pressure loss, efficiency, effect of fuel properties, carbon formation; progress and trends in chamber design; development of specific chambers; ignition; selection of materials for flame tubes. Charts, drawings, photographs and references are included.

74–56 "Prestressing an Ultra High-Strength Steel to Perform Even Higher Duty"—Jerome W. Kaufman; Metal Progress, V. 69, n. 5, May, 1956, pp 87–90. Mechanical prestressing of a bomb shackle hook eliminated failures. The steel used in the new hook was said to be heat treatable to

Abstract Number *Abstract Title*

280,000 to 300,000 psi. The new hook was of the same design as the old one that had failed. This mechanical prestressing permitted replacement of an unsatisfactory part without changing its design.

75–56 "British Free - Piston Engine Development"—The Motor Ship; V. XXXVII, n. 432, March-April-May, 1956, pp 16–18. A 1,000 s.h.p. free piston engine has recently been placed on test. The test bed is arranged to permit testing of as many as four free-piston gasifiers with gas turbines and auxiliaries, complete machinery for a 4,000 s.h.p. installation. A proposal for a whaling vessel includes two free-piston gasifiers (GS-34) supplying gas to a single non-reversing gas turbine arranged to drive a controllable-pitch propeller through reduction gearing. This is a 2,000 s.h.p. propulsion plant. This proposed plant would replace a steam plant, 1,850 s.h.p. triple expansion. The free-piston plant is 20% lighter than the reciprocating steam plant.

76–56 "How Graphite Performs at High Temperatures"—L. D. Loch; Materials & Methods, V. 43, n. 5, May, 1956, pp 126–129. Artificial and natural are the two general types of graphite products. Natural graphites have limited load bearing capacities. The artificial graphites are finding uses in applications where high temperature stability is essential. Graphite is stronger at 4500°F than at room temperature. It also has higher thermal conductivity than other refractories and low thermal expansion. Applications include: dies, molds, crucibles, rocket motor guide vanes, rocket nozzle sections, moderators in nuclear reactors. Graphite offers promise for solution of some high temperature problems in fields of gas turbines, rockets and nuclear power. Curves are presented on tensile strength, creep rates and thermal conductivity. References.

77–56 "Applying Radiation in Industry"—Nucleonics; V. 14, n. 5, May, 1956, pp 41–72. The industrial applications of radioisotopes are presented in this special report. It is estimated that there are now some 1400 industrial groups applying, routinely, radioisotopes to their problems involving research, production, analytical and process control. It has been estimated that industry saved $150-200-million by use of radioisotopes in 1955. The annual saving may increase to $1 billion in the next decade. The report includes the following:

1. "Radioisotopes—Versatile Research Tools"—G. D. Calkins.
2. "Measuring Adsorption of Cosmetics"—M. F. Nelson, Jr.
3. "Activation Analysis — What it can do for you"—G. W. Leddicotte.
4. "Activation Shows Trace-Metal Segregation"—G. W. Leddicotte.
5. "Measuring Trace Elements by Activation Analysis"—Robert C. Plumb.
6. "Isotopes Solve Mixing Problems" — D. E. Hull, B. A. Fries, J. G. Tewksbury and G. H. Keirns.

Abstract Number *Abstract Title*

7. "Mixing in Surge Tanks"—D. E. Hull, B. A. Fries, J. G. Tewksbury and G. H. Keirns.
8. "Industry Fights Wear With Isotopes" — M. Eugene Merchant, Hans Ernstand, E. J. Krabacher.
9. "Tool-Wear Tests Speeded" — M. Eugene Merchant, Hans Ernstand, E. J. Krabacher.
10. "Improving Quantitive Analysis"—Charles Rosenbaum.
11. "Measuring the Active Ingredient in an Insecticide"—John T. Craig.
12. "Radiation Measures Density" — Philip E. Ohmart.
13. "Designing a Beta Gage to Control Cigarette Density"—C. S. McArthur.
14. "Industrial Thickness Gages" — George B. Foster.
15. "Following Large Scale Processes" — V. P. Guinn.
16. "Tracing Catalysts in Refinery Crackers"—V. P. Guinn.

78–56 "Metal Spraying with Rockets"—J. W. Buffington and J. A. Browning; The Welding Journal, V. 35, n. 5, May 1956, pp 468–471. A new method of metal spray coating has been devised. This consists of feeding a metal filler rod into the high-temperature, high-velocity jet of a small rocket. This jet is produced by the combustion in the rocket. The filler rod is melted and atomized in the jet into a fine spray of molten particles. This is a new process that is untried by industry. It is considered to have certain advantages over present air-blast guns.

79–56 "Heat Transfer Problems Solved by Electrical Analogy"—R. O. Smith; Aero Digest, V. 72, n. 5, May, 1956, pp 21–24. The heat transfer in a cooled radial-flow gas turbine rotor was investigated by constructing an electrical model with properties similar to the thermal body. Temperature fields in the rotor were computed by first working out a solution in terms of electrical quantities and then transposing them to thermal quantities. The temperature field in a body must be determined to compute the thermal stresses. The similarity between the thermal and electrical systems is explained. Method of computing heat transfer in a body is described.

80–56 "Plastic Springs"—National Bureau of Standards Technical News Bulletin, V. 40, n. 5, May 1956, pp 70–71. Plastic springs can now be mass produced. A practical procedure for this work has been developed by The National Bureau of Standards under sponsorship of The Army Ordnance Corps. Plastic springs are molded by drawing resin soaked glass fibers through copolymer tubing. This tubing is then wrapped in a helix around a mandrel and cured in an oven. After curing the tubing is removed, leaving a solid plastic spring reinforced with glass fibers. These springs should be of value for applications in corrosive atmospheres and in salt air. They have a strength to weight ratio greater than some metals and this may be another advantage.

Discussion of

Half a Century of Progress in Boiler Water Treatment by Frank E. Clarke, JASNE, February 1955

CAPTAIN HENRY C. DINGER, USN, Retired

This comment was written over a year ago and laid aside. I have just read Mr. Clarke's recent article on gas in the February 1956 issue of the J.A.S.N.E. This article excites my commendation. It is, of course, of more recent development.

HAVING READ Mr. Clarke's article on Naval Boiler Water Treatment in the February issue of the Journal I notice that my name is included, but in a somewhat erroneous manner. Also, I am not in agreement with some of the comments covering the period up to World War II and would like to discuss these so that a better understanding of the situation may be possible.

To begin with, while on duty in the design division of the Bureau of Engineering I initiated the order for the tests carried out by Captain Lyon, which probably was early in 1909. Previous to this I had served in the Engineering Departments of the *Columbia, Celtic* (supply refrigerator ship), *Monadnock* and *Marietta* (first installation of B and W boilers in the U. S. Navy). I made a report of the *Marietta's* performance which was quite effective in securing further installations of B and W boilers instead of Niclausse, which were being forced upon the Bureau by political pressure. I also was chief engineer of the armed cruiser *Maryland* and the battleship *Indiana* (1905-1908). The *Indiana* then had eight B. and W. boilers; four with superheat and four without. I suggested and ran a 24-hour test run which demonstrated the superiority of superheat, and we had no appreciable corrosion. There was no feed heater till I converted one of the auxiliary condensers, but her original condenser tubes were intact for twenty years or more. We did have dirt and oil deposits in the boilers.

I also served deck duty on the *Brooklyn, Yorktown, Castine, Marietta, Portsmouth, North Caroline, Montana* and *Nebraska*, and was in Command of the destroyer *Drayton* of the first oil burner group. While serving deck duty I also was familiar with boiler conditions, and published the handbook "Care and Operation of Naval Machinery" (1908). This covered considerable comment on air and oil in boilers feed heating, and the use of soda.

The undated memoranda referred to on page 12 is probably a copy of one of my various recommendations in reference to better care of Naval boilers, and it may have gone back as far as 1903. The above service, I believe, made me familiar with Naval boiler operation of the time. Reference is also made to the *New Jersey* on page 12, and this vessel had B. and W. boilers like practically all other capital vessels of that date. I do not recognize the term, "small drum boiler"; "cross drum" was the type. At this time the Bureau had no plan nor comprehensive instructions for boiler feed water treatment. The Naval Regulations required that the water be distilled and kept slightly alkaline, but did not say definitely with what. This was left to the judgment of respective engineers. Soda ash was generally used and saturation was taken by a hydrometer; alkalinity with litmus paper. Many of the large vessels had small testing equipment to make alkalinity tests and only distilled water was used for boiler feed. However, shore water was used in an emergency.

During this period I never used lime, but employed sal soda instead. I also used other alkalis that were recommended by competent authority. This I did mainly to find something that would take care of the oil and dirt; since there was not much salt scale except when there were condenser leaks. I did not discover anything in particular except that tri-sodium phosphate seemed in some way to give

promise. Loofa sponges were discovered and they did a fairly good job. On the *Maryland* I toyed with the idea of having a float in the feed tank which would have an adjustable surface blow, and which had a flexible tube to the outside. However, this did not work, mainly due to agitation of water level. During the period 1900-1911, I used soda ash and tri-phos, and also hydraulic turbine boiler tube cleaners. These I introduced to the Naval service when I was Chief Engineer of the *Marietta*, which was about 1902.

The use of tri-sodium phosphate for feed-water treatment antidates Hall's patent by many years; but of course his tests indicated how to use and control it to a better advantage. In service he used a special patented compound called meta-phosphate. Hall has not used the same formula for every plant, but he adjusts it to suit temperature, pressure and conditions.

The comments in the upper left of page 13 are not very clear. About the two most well known firms engaged in boiler water treatment in the country were Dearborn Chemical and Bird-Archer. My idea was that some useful information might be gained, but I have no recollection of having ever recommended that the Navy adopt proprietary compounds. The idea was that, from information and tests, more definite knowledge might be gained from which suitable formulas could possibly be developed. This was the same procedure that both Lyon and Solberg later utilized.

During 1905-1908 I was Chief Engineer on both the *Maryland* and the *Indiana*, and I have no recollection of receiving any direction not to use lime, but I believe its use for High Pressure boilers was properly discouraged.

I have no knowledge of Table I and the article does not describe it . . . I am in the dark. However, I would think that the remedy for mud would be a good wash-out. It is possible that some one may have used the term "blow-down" for the process of emptying, instead of the usual process of opening bottom blow for a short time to blow out any sediment periodically.

In reference to Admiral Melville (1907) and the report on the *Missouri*, there is a question in my mind, since Admiral Melville retired in 1903 and had no connection with the Bureau after that.

Incidentally, I served in the Bureau's Design Division for a short time before Admiral Melville retired in 1903, and was also on the *Missouri* during her acceptance trials. Her boiler tubes discharged into the boiler drum above the water line and this made the tubes very susceptible to air corrosion.

There was no Admiral Bacon in the U. S. Navy, but he may have been British, and I fail to see why he is quoted.

In 1907 there was some difficulty in cleaning deposits from the tubes and this was generally accomplished by turbine cleaners. The later warning against mechanical cleaning was due to the fact that the turbine cleaner, if used excessively or unintelligently, removed some of the metal as well as scale. In regard to the article on page 15, "A Little Oil A Good Thing", I cannot agree that any Naval Engineer ever wanted oil in his boilers; or any scale or air either. (See my book of 1908).

Referring to "Corrosion Curbed"—page 16—it is my belief that there was something wrong with the data or construction of these curves, since this indication was not borne out by results in service. Some of this may be due to the fact that small laboratory tests do not accurately reflect the results secured in actual boilers, particularly for feed water. The author also comments on the small number of points used in making these curves. I doubt whether any point of absolute no-corrosion was arrived at, but it may have been approached. The Lyon report did not consider fully the effect of air nor the possible difference between laboratory data and boilers in use, and the fact that he used some small test boilers does not change this. Also, the circulation of the water is not the same in different boilers and it varies with the rate at which the boiler is operated. Although the differences may be slight they are still there, and even identical boilers behave differently in actual service.

The notes referred to on page 20 were prepared by myself in order to give the best up-to-date information on marine water treatment. This information was not published at that time by the Experiment Station since the original intention was to submit it for publication in the Journal of the American Society of Naval Engineers. However, it wound up in the notes, and in preparing these I received letters of information from firms handling boiler compounds and also from most of the United States shipping companies. Mr. O'Neil—Head of the Chemical Laboratory—and Lt. Solberg, who was then my test officer, were both consulted. No laboratory tests were made and no direction were given for operations since I had no such authority; and I considered the Bureau's instructions at that time to be about as good as any then in existence. However, the alkaline requirements were too low for practical use and it would be better to use a larger percent of trisodium phosphate. I do not see where these notes unwittingly admitted that the experts were confused about feed water treatment, and I would say that perhaps the situation still exists even more so. Naval feed and distilled water is soft, and so is the water available at New York and many other ports, so I cannot see how any adverse comment would apply. Neither did I discuss the advantage of lime, but merely stated how it was used and what it might do, along with the description of other materials. I never used it myself, but I understood, and know, that it gave satisfactory results on low pressure boilers and was then used ashore to a considerable extent. Neither can I admit that the warning against excessive blow-down and unnecessary turbining of tubes is poor advice. This was to avoid leaky blow

valves, loss of heat and leaky boiler tubes. Sometime about 1900 the practice of boiling out boilers with a strong alkali solution to remove dirt and oil deposits came into use. The use of potato peelings, starch and perhaps seaweed had anti-dated this and, when properly done, avoided the use of mechanical cleaning, but also did a job that excessive blow-down did not do, in many cases. The use of boiling out was and, I believe, still is a rather important matter in keeping boilers clean. A knowledge of this situation may also show that the Bureau's warning against excessive mechanical cleaning and blow-down may have had some justification. I have known marine boilers—about 1930—kept so clean that no blow-down was used or needed during a year of service.

The author also notes that the no-corrosion curve was constructed from only three points. I knew Captain Lyon very well and conferred with him in 1911 upon his testing procedure when I was at Annapolis in command of the Destroyer, *Drayton*. The boilers of the *Drayton* were then being treated with a mixture of sal soda and tri-sodium phosphate, and they had neither deposits nor corrosion, except for a little pitting near the open end of the short internal feed pipe. This definitely indicated air corrosion, and was remedied by installing a long pipe with small holes and filling pits with iron cement. I was not on engineering duty at the time Captain Lyon submitted his report, nor was I present when the Bureau issued its orders concerning feed water treatment, since I went to the *Utah* as its chief early in 1912.

About that time various vessels began to find some corrosion in boilers as well as severe priming, which is something I never had any real amount of on any boilers under my control, and very little corrosion. Originally the Babcock boilers on the *Utah*, as well as other vessels of that date, had a feed nozzle by which the feed was discharged above the water level at one end of the steam drum, originated by English B. and W. so that the feed coming in released its air in one place. The boilers had been treated with the high alkalinity feed water regime from the time of her commissioning, and this experience, along with others on other vessels definitely showed that the high alkalinity formula did not prevent corrosion in all cases.

While on the *Utah* we installed a long metal perforated internal feed pipe, surrounded by a sheet metal trough, which was not adopted by the Bureau and was considered too complicated. If used it would have taken care of air corrosion in boilers of that date, but the idea was that any released air from feed water could corrode the sheet metal; thus protecting the boiler drum and tubes. The soundness of this idea was proved in Yarrow boilers circa 1918, feed water box in lower drum to feed heat. The metal of the box corroded on the boilers where its water was not properly treated, but not on others. Because of this trouble the Bureau eliminated the feed box on Yarrow boilers, which was a concession to bad operation at economy's expense. However, I installed a feed baffle in a boiler steam drum at the station which lowered the uptake temperature by 50°F at moderate rates. This sheet baffle did not corrode, and if it did it would protect the tube sheet. An official test of this baffle, the cost of which was probably not over twenty-five dollars per boiler drum, was authorized at Philadelphia, but was not made or was delayed due to low priority. About this time there was considerable gossip among engineer officers as to who was responsible for the formula; was it Lyon or some officer in the Bureau? One story was that when the instructions were being prepared the bureaucrats and Lyon had some difficulty in agreeing, and that the Bureau's ideas prevailed—hence high alkalinity, but not as high as Lyon wished. I made contact when I became Chief Engineer of the *Utah* in 1912.

This vessel had used Navy Compound from the start, and also had a large air separator at the highest point of the feed line. The idea was that any air in the water might be collected in this unit and could be drawn off by an air-cock. This was probably the first installation of any kind of an air separator connected with Naval Boilers and it did not take all the air out of the feed system, but it did take some. Other air was removed by a hot feed tank. (See my book)

Soon after I arrived on the *Utah* an examination of the boilers disclosed extensive corrosion under the feed nozzle on the drums and also a small amount on the headers. Similar corrosion was found on other ships. Nozzles were removed and long internal pipes installed. After this the treatment was more carefully carried out and the alkalinity held at a higher point. Soon after that our brass blow-down valves started to pit and corrode, and it was with great difficulty that they could be kept tight. Other brass fittings on the boilers were also acted upon, and reports of the situation were made from the *Utah* and other vessels to the Bureau. Some vessels reported priming and difficulty in maintaining the required alkalinity, but the Bureau did not act; so I finally took one of the corroded valves up to the Bureau and told them what was happening. Soon after this (1912) the Bureau made the change referred to on page 18. After this we could keep our valves tight and had no appreciable corrosion. The ship also won the Engineering trophy by a large margin. Clean boilers inside and out were a big factor, and we also kept our feed tank hot by waste drains. However, I would not claim that the treated water was not useful for any other purpose because we fed treating chemicals continuously into feed tank, and when any boilers got out of line they were given individual doses.

We also pretreated the fresh water reserve tanks to some extent and this made it easier to keep the boiler water at a more even alkalinity. Then too, it protected these tanks from corrosion and did not re-

quire cement; we also DID NOT cement or coat our water tanks with bitumastic, nor did other Naval vessels at that time.

There is no question but what Captain Lyon's tests gave useful information, but they did not give a point of no-corrosion when used in seagoing boilers. Also, when the Bureau changed to lower alkalinity they went somewhat too low, and during 1926-1930, while in command of the Experiment Station, service reports of boiler corrosion and casualties showed, in most cases, that troubles were due to the fact that the alkalinity was carried too low. I then repeatedly recommended to the Bureau that the lower limit be raised to insure that the water in all parts of the boiler be definitely alkaline at all times.

While the 1911 formula provided means for control it did not prevent corrosion in many cases, and one patent reason for this was that it did not take into consideration the varying presence of air in the water, together with changes in temperature and circulation.

A formula that works well for a fire tube 150 pound boiler does not necessarily give complete satisfaction in a small tube boiler with 1,000 pounds of pressure. Another thing questionable was the small percentage of tri-sodium phosphate. Later on Solberg went to disodium and Hall went to his patented meta-phosphate, while others went to Colloids, Kutch, sodium alginate, etc. Good results were secured when properly carried out, but the author also seems to be of the erroneous impression that Captain Lyon exercised some influence over the control of the treatment after his report was made. His influence was limited to the Journal article and remarks in the Boiler Book, and I doubt whether he had personally much to do with the later revisions of the Boiler Book.

In connection with the letter to Commander Paine —page 21—the Station was only in a position to make a chemical test of the compound and to comment from general knowledge of the art, along with recommendation of a service test. Hence my remark to Commander Paine. At this time there was pressure from various proprietary compounds.

Also, what is so strange about who was to pay the expense of the test? They had to be paid for and available money was very scarce.

The author is also in error when he surmises that Hall and Partridge inspired the Navy to take a new look at its boiler-water treatment. Hall was trying to have the Navy adopt his proprietary system for a price, and at that time I did not consider Hall's system superior, provided alkalinity was raised and more tri-sodium phosphate used.

While in command of the E. E. S. I made numerous official recommendations which, if quoted, would have given better information than quotations from private letters and undated memos.

The type of deaerator illustrated on page 21, actually rendered good service, but, no doubt, improvements were possible. I used feed water heaters and feed tanks as far back as 1905 to remove air, and described how to do it in my book.

The remarks on page 28, concerning cement, seem to me to be rather strange. Did it require a boiler test to show that cement was not a desirable additive?

While both soda ash and tri-sodium phosphate are used in successful treatments, each, when used without the other, have also been successful.

Were not the special compounds for the *Wyoming* —page 29—about the same thing as the statement I made in the notes on page 20 which were so highly criticized? The statement, in other words, is that "no one formula is the best for all boilers operating under different conditions." This is true now as well as it was 40 years ago, and in 1927 I believe there were no Navy boilers above about 300 lbs. pressure, except the 600 lb. boiler at the Experiment Station. While at the station I built a 1,000 lb. experimental boiler for use in testing high pressure valve fittings, etc. It had the first 1,000 lbs. welded boiler drum in the United States.

The use of some special chemical to take care of a special case of corrosion is not poor advice. However, the author is now stating that special mixtures are being used on some of the latter vessels. Also, the private treatment companies vary their treatment to suit pressure, circulation and source of feed water. In view of this I do not see any excuse for the word "inconsistency," etc. Also, there was no statement that the chemicals in Table 8 were to be sent to a ship, or ships. Such chemicals were being used on marine boilers and water glass was used successfully on evaporators to hinder scaling. Solberg was favorable to this and used it at sea. The paragraph "conclusions", etc., was not a conclusion but only a statement covering actual experiences.

In connection with Paine's letter—page 21—This was not an official letter and it might be asked what was the connection with the *Saratoga*. This is it: The boilers of this vessel before delivery were in bad shape due to the fact that shore water, without treatment, was used while in the builder's custody. As a consequence tubes were dirty and corroded. Hence it was a very good place to try out any promising treatment.

The main difficulty with the first Navy Boilers Compound may be summarized as follows: (This was the Lyon report as modified by the Bureau); that is: a somewhat lower alkalinity, which was issued by Bureau officers who did not possess much familiarity with the subject. This time was a period when the Bureau was being run partly by non-engineers or perhaps by some with limited engineering experience.

1. Lyon's tests were not made with actual sea water but with fresh water and Severn water, which had oysters, crabs and vegetable matter in it, making it more or less undeterminable as well as variable. So when the compound was used on actual

Navy ships, in many cases it did not work as expected.

2. Due to high alkalinity, priming was caused, brass boiler fittings and bottom blow valves were rapidly corroded. Some scale was removed but corrosion was also found.

3. Lyon's tests did not take fully or adequately into consideration the effects of air on the boiler. The other thing was that a too small percentage of tri-sodium phosphate was used and some tannin (both of these have some effect as air scavengers). Since steam pressure was going up it would have been better to have increased the percent of tri-sodium since the latter is also a very efficient cleanser and would help to remove soft deposits. The superior effect of Phosphate was what Hall got his patent on, but phosphate had been used in Navy and Marine boilers many years before. It was used by guess but it worked.

4. The so-called three-point-non-corrosion curve did not really show such a point. In practice on seagoing ships it was found that alkalinity could be carried much lower with much better results with the pressure then encountered.

5. After the trouble with the high alkalinity the Bureau changed to low alkalinity in 1913 and then they went too far in the other direction. In service, due to a tendency to maintain low alkalinity, some parts of the water tube boilers had *no* alkalinity. A slightly higher alkalinity would have insured all parts of the boiler to have a definite alkalinity and would also keep the boiler tubes cleaner. Since there were boilers of different pressures, circulation and other characteristics, the alkalinity should have been high enough to secure suitable alkalinity in all parts of the boiler and in all Navy boilers.

After coming to the Engineering Experiment Station in 1926 I found no technical force there having charge of feed-water treatment and, in fact, the Station did not use Navy treatment in its own boilers. However, boiler casualties were referred to the Station. These involved burned-out tubes as well as corrosion due to improper feed water. After a number of such reports were received I could see that the real cause for most of the trouble was that boiler water treatment had not been used properly, or the alkalinity was carried too low for the conditions. I then started to recommend to the Bureau to raise the lower alkalinity requirement so that a safe alkalinity would be present in all boilers and in all parts of each boiler. The Bureau did not act on this, so I started, at this time, to compile the notes on feed water treatment which were published in notes for the J.A.S.N.E. This was a summation of the then existent information including merchant fleets. These bad-advice notes received rather favorable comment in Britain.

It was not a directive to the ship's personnel to regulate feed water treatment, so most of the comments regarding this paper become more or less ex parte' and unfortunate. The statement in the notes that ships trading in different areas and up various tropical rivers would find that a variation in chemical content might be desirable, and a formula suitable for 250 lbs. pressure, might not be suitable for six hundred, was a fact, and the Bureau's idea on this was that there would be one standard compound for all boilers so that there would be a standard supply that could be sent all over the world. This was practical when pressures were below 300 lbs., but when higher pressure came the Bureau could have authorized the high pressure ships to use certain additives, such as phosphate and air scavengers, so as to keep these newer vessels on a formula that was suitable for them. While at the Station tests were authorized for Hall, and also some others which involved sodium alginate and other so-called colloids. Some of these showed up very well, but the matter of chemical control was uncertain. It would not be very practical to hire any of these companies to operate the Navy feed water treatment all over the world. It might have been practicable to hire them for a limited number of vessels or to purchase some of their special compounds. I might also mention that in 1934, when Commander Solberg delivered his lecture at Mechanical Engineers, New York, the private feed water representatives were quite critical of the use of the new Navy Compound and some unreasonable predictions of doom were heard. As a matter of fact, the new formula was not much different from the old except that disodium was used instead of tri-sodium, tannin taken out and alkalinity better regulated to respond to the use of higher pressure. I also believe that the Solberg tests failed to corroborate Lyon's point of no-corrosion. It is not one point, but a space line beginning with a margin of definite alkalinity and ending where the alkalinity becomes so strong that it will attack the metals that contain it. This varies for different metals, pressures, and various and sundry ingredients in the boiler water, which the operator often does not know about; for instance, oxygen in various forms. Neither did the Solberg formula fully cover control of air in high pressure boilers except possibly advice to keep it out. This is a precept older than the Bureau of Steam Engineering.

I am sorry that Mr. Clarke has relied on undated memos, unofficial letters and not more on official papers, or action, so his reference to my papers become somewhat inappropro. In particular, the apparent humor concerning the fact that I was supposed to have asked Commander Paine to whom a proposed Hall test was to be charged. In my day any tests made for contractors or private firms were paid for by them.

It was only when the Bureau initiated some test that Government funds paid.

In my day the station had a very small appropriation and it had to be regarded carefully. It was only by saving fuel, machinery, scrap and considerable junk, selling this at auction or turning it over to Norfolk or the Washington Navy Yard, for which we

received credit, that I was able to increase the working force by about 20% and also to promote a certain number of the employees every quarter. This policy paid well in results. Without increase in appropriations I was able to start the new laboratories for testing roller bearings, diesel engines and preliminary work for welding, and to get the data on corrosion fatigue into some shape so it could become useful to the Navy and to industry, as well as to correct some of the extreme claims then being made for the theory of caustic embrittlement.

Mr. Clarke, unfortunately, appears to be handicapped by the fact that he has endeavored to describe a situation with which he is not personally familiar and concerning which reliable data is very meager. Hence assumptions are made which in some cases do not well reflect the actual situation and even bring some of the authorities unjustifiably into ridicule. Another strange thing to me is that "Speller" is not referred to in the paper. He was the first outstanding practical authority on metal corrosion in the United States.

I have written this comment from memory, have consulted the Lyon article and re-read the bad advice notes published in the J.A.S.N.E. However, my memory is quite clear on this matter having lived with it almost continuously for something over 50 years.

Comments by Frank E. Clarke on Discussion by Captain Henry C. Dinger of Article "Half a Century of Progress in Naval Boiler Water Treatment"

Captain Dinger's discussion is a very interesting and detailed account of the early history of Naval Water treatment by one who was both an eye witness and a recognized early authority in the field. Several parts of the discussion merit further comment and explanation.

My discussion of Captain Dinger's part in Naval water treatment undoubtedly was incomplete, as the records were incomplete. Thus, incomplete would be a better description of the deficiency than "somewhat erroneous".

The text of Page 13 of the basic article is essentially the wording of the official Navy record. It does not recommend use of proprietary compounds, but does show that Captain Dinger was using the scientific approach, as he recollects.

Admiral Melville did retire before 1909. The ASME notes of that year cover an address by him to the American Society of Mechanical Engineers on the subject "The Modern Navy — An Engineering Affair". This same series of notes covered comments on boiler care by Captain Dinger from International and Marine Engineering. Because a title page was missed in the original search of the Station's well used 1909 ASME volume, the comments attributed to Admiral Melville actually were made by Captain Dinger.

Admiral Bacon was British. His ideas were included in the article to show the thinking of the day just as references were made to industrial progress throughout the article.

The notes on Page 20 of the article were recognized as a compilation of opinions, hence the plural "experts". Reference to "inconsistencies" was not a serious reflection on 1920 science. Captain Dinger certainly is right in concluding that confusion still exists, and it always will.

While Captain Dinger did not use cemented tanks some operators did. The official records contain formulas for them and the author encountered them personally as late as the 1940s.

The assumption that the work of Hall and Partridge inspired further Navy work was based on the fact that Dr. Partridge's assistance was requested in planning the research program of 1931.

Comments on Naval correspondence regarding trial of the Hall treatment was not intended to be humorous and definitely did not intentionally contain ridicule in any sense. They simply expressed surprise at the existing policy regarding proprietary products and the apparent shortage of research funds in a time of reported prosperity.

The author's personal friend Frank Speller was not mentioned in this article because there was no mention of him in the Navy records. He certainly was, and still is, a recognized authority on corrosion.

The author too, regrets that memoranda and letters were the only link in some of the evolution described. Unfortunately much of the "official" correspondence referred to by Captain Dinger has disappeared. It definitely is a handicap not to have experienced this history personally. This problem is common to all historians.

BOOK REVIEWS & NOTICES

JET ENGINE MANUAL

by

R. Mangham and A. Peace

Published in 1955 by

Philosophical Library, Inc.

15 East 40th St., New York 16, N.Y.

Reviewed by W. H. Cullin, CDR, USN

The object of the book, as propounded by the authors, is to present information, in a question and answer form, for use by men engaged in the operation and maintenance of gas turbine aircraft engines.

The book covers the construction, mechanical and functional details, fuel, oil, and starter systems, controls, operation and testing of English turbojet and turboprop engines with additional chapters on installation, fault diagnosis and component inspections. As the aforementioned list conveys, the coverage of Aeronautical Gas Turbine Engines is rather complete. However, the price of $3.75 per copy for 128 pages of information is considered too high.

It is also considered by this reviewer that the book fails to meet the mark as far as U.S. readers are concerned, because the engines concerned are English engines, and therefore the methods of construction and component detail are different. This difference is enough to invalidate a substantial portion of the book.

A glaring deficiency from a U.S. user's standpoint is the fact that air turbine starters are not discussed at all. This type of starter is in use on all newer, big U.S. turbojet engines. On the other hand only one engine, the J65, the English Sapphire engine converted for American manufacture, uses the cartridge starter. The newer big turbojets would require a cartridge almost the size of a 5-inch shell to accomplish the job.

English terminology is used in critical areas which limit the effectiveness of the descriptive material, i.e., high pressure cock, fuel trimmer, nip between blades, barostat, etc.

In general, the coverage of systems is good but the background counterpoint that detracts from the authority of the authors is that the component, situation and/or condition being described, is tailored to a specific engine and/or component that is obviously English, e.g., "the governor screw requires to be screwed in to increase rpm and is locked by a cap nut." The answer to the question "How do the fuel pump and barometric pressure control interact when altitude is increased or decreased" is a description and explanation of a specific unit which is not applicable, in general, to all U.S. engines. This information is readily available in specific engine handbooks. The descriptions and explanation, in a great many cases, are apparently of specific components which are not identified *per se* and therefore, the material is suspect. Thus the book becomes a dubious high-priced maintenance manual of information that is normally available from the engine manufacturer.

However, the final chapter on fault diagnosis and component inspection—the authors mean inspection in a preventive maintenance sense—is excellent and full of good common horse sense, and is the best feature of an otherwise mediocre book.

Comments

by

VICE ADMIRAL HAROLD G. BOWEN, U. S. Navy, Retired

the Author, on the Review of "Ships, Machinery and Mossbacks" by Geoffrey Cumberlege for the Annual Steam Number, 1955, of the Marine Engineers and Naval Architects, London, England, which was reprinted on page 401 of the May 1956 issue of the Journal.

This review is thoughtful, kindly and understanding. There are a few criticisms, however, that seem to merit consideration.

MR. CUMBERLEGE STATES: "The old ship-and-engine builder was gone around 1933, the author tells us, for they could no longer compete with the specialist designers and builders of turbines, gears and boilers. It is interesting to contrast this with today's practice in Britain, where we successfully use both types of firms."

COMMENT: I said in my autobiography on page 124, "The shipbuilders never could have supplied the quan-

tity of machinery required by our shipbuilding effort in World War II without General Electric, Westinghouse, and Allis Chalmers. I brought the Foster-Wheeler Corporation into the supply field because Babcock and Wilcox were our only purveyor of boilers, and I wanted to broaden the base of supply for our war building program.

"The same reason applied to my bringing National Battery Company in so that, when the time came, they could supplement the efforts of "Exide" in producing storage batteries for submarines xxx." Similarly, in the submarine engine field, General Motors, Hoover-Owens-Rentschler Company, and Fairbanks Morse Company were all brought in as suppliers of submarine Diesels. England had no such building program in World War II and, therefore, the situation in England and in this country was not comparable. The American system of making assembly yards out of the old ship and engine builders is also responsible for our successful revolution in engineering. I am not aware that the English system produced comparable results.

MR. CUMBERLEGE: "We cannot, however, let the forthright Admiral's statement that the U. S. Navy and not the Royal Navy pioneered in this great advance in naval engineering pass, without reminding him that the British Thornycraft built destroyer *Acheron* had high-pressure, high-temperature machinery in 1928-1930 but she did not complete her trial until 1931."

COMMENT: Let us see how our friend Mr. Noah Webster defines the term "pioneer." "One who goes before, as into the wilderness, preparing the way for others to follow." The *Acheron* experiment pioneered nothing, and Mr. Cumberlege, the reviewer, acknowledges that the "costly and painful lesson which she should have represented to the cause of naval engineering was largely lost with her". The *Acheron* incident is nothing to brag about, it is something to cry about. What on earth were the old line ship and engine builders in England thinking of and why did the Royal Navy draw a blank? Our success with high-pressure, high-temperature and the new engineering revolutionized marine engineering practice the world over, whereas the *Acheron* experiment was sadly a dud. In view of the above, I think, with all respect to the reviewer, my statement must stand — the statement that "the U. S. Navy and not the Royal Navy pioneered in this great advance in naval engineering."

*i.e. The Big Three

Parson's Marine Steam Turbine Co.

THE REVIEWER STATES: "Whether or not the author intended it, the impression is left that the Parson's Marine Steam Turbine Company controlled the machinery design policy of the U. S. Navy from England prior to 1933."

COMMENT: If the reviewer had ever lived in a country that was once a colony, he would have discovered that liberation from the mother country comes by degrees, and is merely begun with independence. In the history of the United States one can observe the progress of their liberation first in one subject then in another, as the colonial mind began to do a little thinking for itself and began making creative contributions. Among our colonial headed it was anathema to criticize either the Royal Navy or Parsons Ltd.

Before 1933, the contractors seem to have dominated our Naval engineering design. These contractors were the Big Three, all of whom were Parsons' licensees. They had an idolatrous faith in Parsons, with the result that in effect it was true that Parsons was in control of marine engineering design in the U. S. Navy Bureau of Engineering. The Espionage Act gave us the opportunity of forcing these yards to do a little thinking for themselves and stop passing the buck to Parsons.

THE REVIEWER STATES that the story of the high-pressure, high-temperature fight "ends rather unsatisfactorily with 'the Board's peroration, saying in effect that they would have none of it' and 'the narrative suffers somewhat from a tailing-off after the heat had been generated and the memoranda flashed a few years before the war broke out.' "

COMMENT: As a matter of fact, the Board's "peroration" had no effect on the universal application of high-pressure, high-temperature steam, together with the new engineering to the U. S. Navy, as we can see from the quotations which follow.

On page 88 of my autobiography under the subhead, *"High-Pressure High-Temperature in Battleships"* reads as follows:

"North Carolina and Washington. The *North Carolina* and the *Washington* (BB 55 and 56) had been allocated on 24 June 1947 to the New York Navy Yard to build because the bids for these ships by private bidders* were believed to be excessive. Shortly afterwards, and before the work on the ships had proceeded too far, I directed that the steam conditions be changed to 600 p.s.i. and 850° F. from 600 p.s.i. and 700° F."

On page 113, the autobiography reads as follows:

"The *King George V* burned 39% more fuel at low speeds than the *Washington,* and the *Washington* maintained a fuel superiority up to high speeds. Furthermore, the cruising radius of the *Washington* was double that of the *King George V* at low speeds, and very nearly twenty per cent greater at high speeds".

"The *North Carolina,* sister ship to the *Washington,* exceeded her designed speed of 26 Knots by 2 Knots. The *North Carolina* and the *Washington,* it will be remembered, were the first battleships to have high-pressure, high-temperature and the new engineering."

The autobiography contains other references to these two battleships.

On page 90, the autobiography reads as follows, under the heading *"The General Board gets into the Act"*:

"Bids were to be opened for more battleships (BBs 57-60) on 2 November 1938. It had been specified that these ships would be built to 600 p.s.i. and 850° F., but this had been accomplished only after much maneuvering and after the usual controversy that attended every effort to improve naval engineering."

"xxx it was necessary to include in the contracts at the suggestion of Bethlehem, a limited responsibility clause which expressed the idea that since the contract included certain advanced features** in engineering, with which the contractors were not familiar, the Government would assume reponsibility for any work involving

**i.e. High-pressure, high-temperature and all of the new engineering.

additional expense which might be incurred by the contractor when it could be proved that this additional expense was the result of 850° F. and was not due to defective workmanship or defective material."

"In the case of Newport News Shipbuilding and Dry Dock Company** the Government actually purchased and supplied the propulsion machinery for the battleship they built under the 1938 program so that Newport News would not be liable for any such dreadful innovation as high-pressure, high-temperature."

On page 111, the autobiography states as follows:

"High-pressure and high-temperature were installed in the Atlanta class of light cruisers, thus marking the entrance of the new engineering into the cruiser type of vessels."

And again on page 124.

"I standardized on 600 p.s.i. and 850° F., extending it to light cruisers and battleships. It is unfortunate that the term "high-pressure, high-temperature" obscures the fact that what actually took place was a complete re-designing of all the machinery that goes into a naval vessel. As a result, all of our designs were standardized and tried out at sea long before Pearl Harbor. I do not know of any Bureau that was better prepared for World War II than the Bureau of Engineering."

Thus, my autobiography clearly indicates that I succeeded in putting high-pressure and high-temperature, plus the new engineering, in destroyers, cruisers and battleships. The White House prevented me from putting it into aircraft carriers, for reasons noted on page 111 of my autobiography.

** Later, builders of SS *United States*.

EDITER'S NOTE: Admiral Bowen's comments on this review of his book were somewhat longer. Certain portions which did not bear directly on the review are not included above.

MARINE CARGO OPERATIONS

By Captain Charles L. Sauerbier, USNR

Published in 1956 by

John Wiley & Sons, Inc.

440 4th Avenue, New York 16, N. Y.

Price $10.50 — 548 pages

This book stresses the team activity of marine cargo operations. It shows the place of each unit—operations manager, terminal superintendent, marine superintendent, stevedore, ship's master, deck officer, etc.—and stresses the need and the methods of effective coordination by outlining the tasks, concepts, and functions in which each member is involved.

There is a very thorough discussion of the objectives of good stowage, together with the means of obtaining these objectives. Included are detailed discussions of ventilation, distribution of weight, initial stability and trim.

The book is also concerned with efficient and economical cargo handling, discussing the advantages and disadvantages of given methods and machines. Included are descriptions of the "roll-on-roll-off" ship and the "container" ship.

One full chapter is devoted to research and development in the field of marine cargo operations, including a full reprint of the research report, "Engineering Analysis of Cargo Handling."

The author is Chief, Cargo Section, Department of Nautical Science, U. S. Merchant Marine Academy.

Chapter Headings:

Shipowner's Organization for Cargo Procurement, Stowage Care, and Delivery. Cargo Responsibility. Principles of Stowage. Planning the Stowage. Stowage of the Cargo. The Ship's Loading and Discharging Equipment. Materials Handling Principles and Equipment. The Ventilation of Cargo Holds. Research and Development in the Industry.

GEOLOGY AND OURSELVES

by

F. H. Edmunds

Published in 1956 by Philosophical Library, Inc.

15 East 40th St., New York 16, N. Y.

246 pages of text and illustrations—$10.00

The author has been a member of the staff of the Geological Survey of Great Britain for more than thirty years.

The purpose of the book is well explained in the first paragraph of the author's preface:

"During the course of his official duties a geologi-

cal surveyor meets people in many walks of life who usually show a lively interest in his work, but who, for the most part, have little or no geological knowledge. Two questions that constantly arise are 'what do you do?' and 'What is the economic use of geology?' It is to the second of these questions that this book is designed primarily to reply, largely in general terms, but in some instances in detail."

Quite naturally British experience and geography predominate where example are used. This will probably give an alien tone to many American readers because it is easy to overlook the fact that the author's knowledge is based on world-wide experience and study even though the things which are closest to him are understandably British.

The knowledge of the author is disclosed by his very interesting sketches of the developments of some of the branches of geology from folklore through art to science. These, although not the central theme of the book add a great deal to its readability and perhaps to the thoughtful reader, will point up the value of the knowledge which the students of geology through the ages have amassed.

The naval reader of this book can, with very little stretch of the imagination, gain a good insight into one of the basic subjects which underlies logistics and Economic Mobilization, those things which in the art of war are beginning to overshadow conventional strategy and tactics. Books like "Geology and Ourselves" are easy and interesting reading which can broaden a specialist in the naval or or any field because truly, the "Ourselves" in the title means just that—each of us.

ATOMS and ENERGY

by

Professor H. S. W. Massey

Published in 1956 by Philisophical Library, Inc.

15 East 40th St., New York 16, N. Y.

158 pages of text and illustrations—$4.75

Of the non-technical works on atomic energy, this small volume is truly that, without leaving in a limited technical mind the feeling of trivial treatment. The author is Quain Professor of Physics in the University of London. During the war he was head of the group of British scientists who worked in this country on the A-bomb project. He is vice-president of the Atom Scientists Association.

Although the relation of atomic research to atomic energy forms the central subject of the book, special attention has been paid to a description of the nature of the research methods used in modern physics. The final chapter gives an account of the researches going on at present to elucidate the nature of the fundamental particles—electrons, protons, neutrons, pi-, mu- and V-mesons and neutrinos—without any reference to applications. The way in which atomic energy is converted to heat in the sun and stars is also described.

SAILING ALONE AROUND THE WORLD

by

Captain Joshua Slocum

Edition Copyright 1956

by Dover Publications, Inc., New York

$1.00

A new popular priced, paper-backed edition of Captain Slocum's story of his solo circumnavigation of the world is now available. This 234 page book provides easy and interesting reading as Captain Slocum recounts very sketchily his adventuresome feat of sailing entirely by himself the sloop *Spray* forty-six thousand miles around the world during an absence from Fairhaven of thirty-eight months. It could not be done. He did it.

Fleet Admiral

ERNEST JOSEPH KING

U. S. Navy

In 1923, while serving as Chief of the Bureau of Aeronautics, Admiral King became the twenty-ninth President of the Society.

Admiral King had been personally selected by President Roosevelt as Chief of Bureau after a career of thirty-seven years. This had been marked by normal, but abnormally diverse, duty assignments in each of which he added to the reputation which led to his last active assignment. As Chief of Naval Operations and Commander-in-Chief, U. S. Fleets during World War II Admiral King became the most important naval commander of all history, by several criteria. His performance places him among the "great captains of history".

Admiral King's death at the age of 78 closed a total naval career of 60 years. Only about a year before his death he dropped into the Society's old office on F Street to chat about "old times" with Mr. Fessenden. Although the heavy responsibilities of his World War II job had taken their physical toll which had slowed him down considerably below his work of the past fifty years, his mind had not lost the sharp acuity which had always kept his associates on the qui vive.

As stated earlier, Admiral King is already recognized as a "great captain of history". His position in history for all time is assured.

IT IS A SAD DUTY TO REPORT THE FOLLOWING DEATHS

DANTZLER, TILLMAN T.
Captain, U.S.N.

KING, ERNEST J.
Admiral of the Fleet, U.S.N.

OLCOTT, FLOYD B.
Civil Member

WELLS, WILLIAM C.
Civil Member

CHANGES IN MEMBERSHIP

ADDITIONS TO MEMBERSHIP

It is with great pleasure that the following additions to our membership since publication of the May 1956 JOURNAL are announced.

NAVAL

Beyer, Stuart Edward, Lieutenant, USN
54 Easton Street, Lowville, N.Y.

Bishop, Michael Edward, Ensign, USN
USS *Vogelgesang* (DD 862)
Fleet Post Office, New York, N.Y.

Brown, William Drane, Rear Admiral, USN (Ret.)
1654 River Road, Apt. 11, Jacksonville 7, Fla.

Eckelmeyer, Edward Herman, Captain, USN
5109 Hampden Lane, Bethesda 14, Md.

Ewell, Roy Holt, Lieutenant, USNR
Merchant Marine Office, Esso Shipping Co.
Mail: 56 Beverly Road, Bellmore, L.I., New York

Fashbaugh, Robert Charles, Lieut. Comdr., USNR-R
Lubrication Salesman, Manufacturers' Representative
7040 North 14th Place, Phoenix, Arizona

Garcia, Carlos Anthony, Ensign, USCG
USCG *Androscoggin* (WPG 68)
Mail: 29 Alhambra Plaza, Coral Gables, Fla.

Garrett, Wallace Henry, Commander, USN
9501 Milstead Drive, Bethesda 14, Md.

Harris, Lyman Charles, Lieut., USN
70 Marble Hill Ave., (Apt. 5-K) New York, N.Y.

Henderson, James Lancaster, Comdr. USN
Supervisor of Shipbuilding, USN, Bath, Me.

Johnson, John Norton, Comdr., USNR
Ten Eyck Associates, Washington, D.C.
Mail: 7 Acton Place, Annapolis, Md.

Jones, Russell Burton, Commander, USN
111 City Park Ave., Portsmouth, Va.

Kashuba, George John, Lieutenant, USCG
4004 The Alameda, Baltimore 18, Md.

Keating, Gerald Thomas, Lieutenant, USN
USS *Bon Homme Richards*
Mail: 1373 Brooklyn Ave., Brooklyn 3, N.Y.

Keller, William Walton, Captain, USN
Rm. 3135 Bureau of Ordnance, Navy Dept.,
Washington, D.C.

Lansdowne, Falkland MacKinnon, Captain, USN
5702 Wilson Lane, Bethesda, Md.

Lee, Paul H., Lieutenant, USNR
c/o Intelec, Edificio Industria Este 2, Pte. Republica,
Caracas, Venezuela

Litten, Glen Allen, Comdr., USN
Inspector of Naval Material, USN
2300 11th Ave., S.W., Seattle 4, Wash.

Loesch, Robert C., Lieutenant, USNR
Sales Engineer, Bailey Meter Co.
1050 Ivanhoe Rd., Cleveland 10, Ohio
Mail: 107 Kensington Oval, Rocky River 16, Ohio

MacKay, George Lewis, Lieutenant, USN
P.O. Box 749, Ocala, Fla.

McAdams, John Whitfield, Jr., Lieutenant, USN
USS *Valey Forge* (CVS-45)
Fleet Post Office, New York, N.Y.

Miller, Bruce Alan, Lieutenant, USN
USS *Chopper* (SS 342)
U. S. Submarine Base, Key West, Fla.

Olsen, Ralph Andrew, Lieutenant, USNR
Chief, Structural Test Facility, Reed Research, Inc.
1048 Potomac St., Washington, D.C.

Perkins, Robert Edward, Captain, USN
Supervisor of Shipbuilding, USN & NIO
118 E. Howard St., Quincy 69, Mass.

Pfeiffer, Arthur, Commander, USCG
Chief Mach. Sec., USCG, D.C.
Mail: 2207 Dayton St., Silver Spring, Md.

Poche, Adolph Judson, Jr., 2d Lieutenant, USMC
1617 Stafford Ave., Fredericksburg, Va.

Scholes, Richard E., Lieutenant, USNR
Greer Mache Corp., 81 S. Bergen Place, Freeport,
N.Y.
Mail: 261-01 Langston Ave., Glen Park, Long Island,
N.Y.

Shook, Loring O., Captain, USN, (Ret.)
310 East 10th St., Cookeville, Tenn.

Uberti, George A., Ensign, USNR
Engineering Specialist, Machinery Scientific Section
N.Y. Shipbuilding Corp.
Mail: 1719 Chelsea Road, Melrose Park, Penna.

Wetzell, Howard Bardh, Lieutenant, USN
USS *Pocono* (AGC-16)
Fleet Post Office, New York, N.Y.

CIVIL

Armstrong, Leonard J., Design Engineer, Wil-Arm Mfg. Co.
Mail: 27 Clove Lake Place, Staten Island 10, New York, N.Y.

Boland, Edward L.
The Babcock & Wilcox Company
161 E. 42d St., New York 17, N.Y.

Cheyney, Charles C., Vice-President, Sales & Engineering
Buffalo Forge Co., 490 Broadway, Buffalo, N.Y.

Cowell, Richard B., Head, Engineering Div. Scientific Dept.
U.S. Navy Mine Defense Laboratory
Mail: 542 S. Bonita Ave., Panama City, Fla.

DeWolf, Loren L., Superintendant Marine Erection Dept.
Babcock & Wilcox Co., 161 E. 42d St., New York 17, N.Y.

Doyle, James P., Engineer, Gibbs & Cox, Inc.
Mail: 300 Morris Ave., Mountain Lakes, N.J.

Fritz, William A., Head, Heat Exchanger Div.
Naval Boiler & Turbine Lab., Phila.
Mail: 517 Saxer Ave., Springfield, Del. Co., Penna.

Harper, John B., Project Engineer, Detroit Controls Corp.
Control Engineering Unit, Norwood, Mass.
Mail: 67 Old Middlesex Road, Belmont 78, Mass.

Hasek, C. W., Jr., Manager, Marine Design & Estimating
Babcock & Wilcox Co., New York
Mail: 318 Libby Ave., Ridgewood, N.J.

Heath, William R., Executive Vice-President
Buffalo Forge Co., 490 Broadway, Buffalo, N.Y.

Hradesky, Edward Lee, Engineer, Co-Ordinator Missile Projects
Loewy Hydropress Div., Baldwin Lima Hamilton Corp.
Mail: Box 597, R.D. 1, Uniontown, Penna.

Mitchell, Harry Francis, Supervisory Mech, Engr. In charge of Design Test Units, Charleston Naval Shipyard
Mail: 10 Vanderhorst St., Charleston 17, S.C.

Panella, Anthony S., Supervisory Naval Archt.
Bureau of Ships, Navy Dept.
Mail: 7415 Dickenson St., Springfield, Va.

Rhodes, Wesley, Project Engineer, Marine Equipment
C. H. Wheeler Co., Philadelphia, Penna.
Mail: 825 S. Cecil St., W. Phila., 43, Penna.

Seneff, Harold L., Chief Elect. Engineer, S5W Project
Electric Boat Div., Gen. Dynamics Corp., Groton, Conn.

Strang, Walter D., Jr., Design Co-Ordinator
Electric Boat Div., Gen. Dynamic Corp.,
Mail: Brown's Crossing, Gales Ferry, Conn.

ASSOCIATE

Albert, Frank J., Chief Engr., SS *William Lykes*
Lykes Bros. S. S. Co., New Orleans, La.
Mail: 4975 Lafaye St., New Orleans, La.

Beebe, Fred, Mech. Engr. Field Thompson-Lightner Co.
O.I.C.C. Officer, Navy 103, Fleet P.O., New York, N.Y.

Lackey, Homer C., Manager Forgings Sales
U.S. Steel Corp., 525 William Penn Place, Pittsburg 30, Penna.

Potter, Harold Clinton, Sales Mgr., Detection Equip.
General Electric Co., French Road, Utica, N.Y.

Pryor, Harry, Sales Engineer
Electric Storage Battery Co., 1819 L St., N.W., Washington 6, D.C.

Veltre, Albert Peter, Mech. Design Engineer
44 Talbott St., Baltimore 25, Md.

Ward, John R., District Manager
Vane Heating Corp., 734 15th St., N.W. Washington, D.C.

JUNIOR

Cushing, Charles R., USMMA, Class 1956
Mail: 187 East 35th St., Brooklyn, N.Y.

Dudziak, Donald J., UCMMA, Class 1956
Room 1202, Palmer Hall
U.S. Merchant Marine Academy, Kings Point, N.Y.

Little, Paul Raymond, Univ. of Virginia, Class 1957
220 Venable Hall, Univ. of Virginia, Charlottesville, Va.

Reichelsdorfer, Peter, Univ. of Wisconsin, Class 1956
428 Wisconsin Ave., Sheboygan, Wisc.

Wiegand, Francis Joseph, Jr., Cornell Univ., Class 1957
216 South Irving St., Ridgewood, N.J.

REJOINED

Ackley, Adolphus Wilson, Jr., Naval Member
380 West Mountain Rd., Simesbury, Conn.

TRANSFERRED JUNIOR TO NAVAL

Farrell, Lawrence Joseph
17424 E. Braiserd St., Pensacola, Fla.

RESIGNED

Rickover, Hyman G., Rear Admiral, USN

ASSOCIATION NOTES

ORIGINAL ARTICLES

The editor of the JOURNAL is most anxious to obtain manuscripts of original articles on subjects of interest to naval engineers. Any manuscript accepted for publication becomes the property of the Society which copyrights it upon publication. Authors are paid from $50.00 to $250.00 depending upon length, interest and professional value. Payment is made at time of acceptance.

The rules for manuscripts are as simple as we can make them.

1. They must be legible (typewritten, double-spaced preferred) since no proof is submitted to authors for correction prior to printing.

2. Single copy only is required.

3. Mathematical formulae used must be set up, preferably by hand printing, so that there can be no question as to symbols, subscripts and exponents.

4. Any drawing submitted must be black on white. A cloth or paper tracing is acceptable but not required.

5 Photographs may be negatives, but glossy prints are acceptable.

6. *Include on a separate page a short biographical sketch(es) of the author(s).* 50 to 100 words for each author is desired.

Manuscripts should be addressed to:

Secretary-Treasurer
American Society of Naval Engineers, Inc.
1012 14th St., N.W.
Washington 4, D.C.

Each author who is subject to the Security Regulations of the Department of Defense, is personally responsible for the clearance of an original article before submitting it to the Society for consideration. In forwarding manuscripts, a statement in this regard should be included.

Manuscripts accepted will not be returned unless specifically requested by the author. If returned, they will be in the condition which has resulted from the work of the printer and the engraver. Immediately following publication, the author is furnished 20 reprint copies of his article free of charge. Additional copies may be purchased from the Society *if the request is received 30 days prior to the publication date which is the 25th of the issue month.* Estimate of cost of additional reprints, which will vary with the nature of the article and the number of copies ordered, will be furnished on request as soon as possible after the article is set up.

Manuscripts not accepted will be returned, postpaid, by the Society.

Manuscripts which are accepted will be published in the earliest JOURNAL which has not yet been closed (60 days before publication) and for which sufficient material is not already on hand.

ASSOCIATION NOTES

PHOTOGRAPHS

Photographs of current or historical interest to readers of the JOURNAL are desired. Any which are accepted by the Society will be purchased by the Society at a standard price of $5.00 each. Such purchase *will not* include any copyright by the Society, but the contributor must hold the Society free from any charge of violation of any previous copyright.

NOTICE OF DEATHS

The Society has no satisfactory machinery for obtaining notice of deaths of members. It is particularly desired that we receive such notice upon the death of any past officer, past member of the Council or member of more than twenty years' standing with photographs and short obituaries. It will be appreciated greatly if any member who learns of the death of a member will advise the Secretary-Treasurer.

SOCIETY LAPEL BUTTONS

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a very fine dignified insignia. It is one-half inch in diameter.

The oak leaves and lettering are red on a gold background.

It is available to all members at fifty cents (50c) each.

PERMISSION TO REPRINT

All material published in the JOURNAL, except the articles which are prefaced with an "Acknowledgement," are copyrighted by the American Society of Naval Engineers.

Permission is granted to reprint any copyrighted material contained herein if the following conditions are met:

a) The article is reprinted in full. Extracts or summaries or condensations may be printed with the consent of the author.

b) Credit is given to the JOURNAL with reference to the issue.

c) Credit is given to the author.

d) If the author is a military officer or a civilian employe of the Department of Defense, the following note shall be carried:

"The views expressed herein are the personal opinions of the author and are not necessarily the official views of the Department of Defense or of a Military Department."

APPLICATION FOR MEMBERSHIP

IN THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.

(See reverse side for required qualifications for various classes of membership)

Date _____

I, _____ hereby apply for membership in the American Society of Naval Engineers, Inc., and enclose $7.50 as my annual membership dues for the year _____ $6.00 of which is for a subscription to the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC., for one year. I submit the following information:

For Naval Membership

Name _____
(First) (Middle) (Last)

Rank _____ File No. _____

Business connection and position, if any _____

For Civil Membership

Name _____
(First) (Middle) (Last)

Years in engineering work _____

Years in responsible charge of important work _____

Present business connection and position _____

Recommended by (two members) _____

_____

For Associate Membership

Name _____
(First) (Middle) (Last)

Rank, if Commissioned Officer of U.S. Army

or of foreign military or Naval service _____

Business connection and position _____

Recommended by (one member) _____

_____
Signature of Applicant

Address for JOURNAL and Mail _____

_____

MAIL TO AMERICAN

MAIL TO SECRETARY-TREASURER
THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.
Suite 1004, Continental Building
1012 14th St., N.W.
Washington 5, D.C.

QUALIFICATIONS FOR MEMBERSHIP

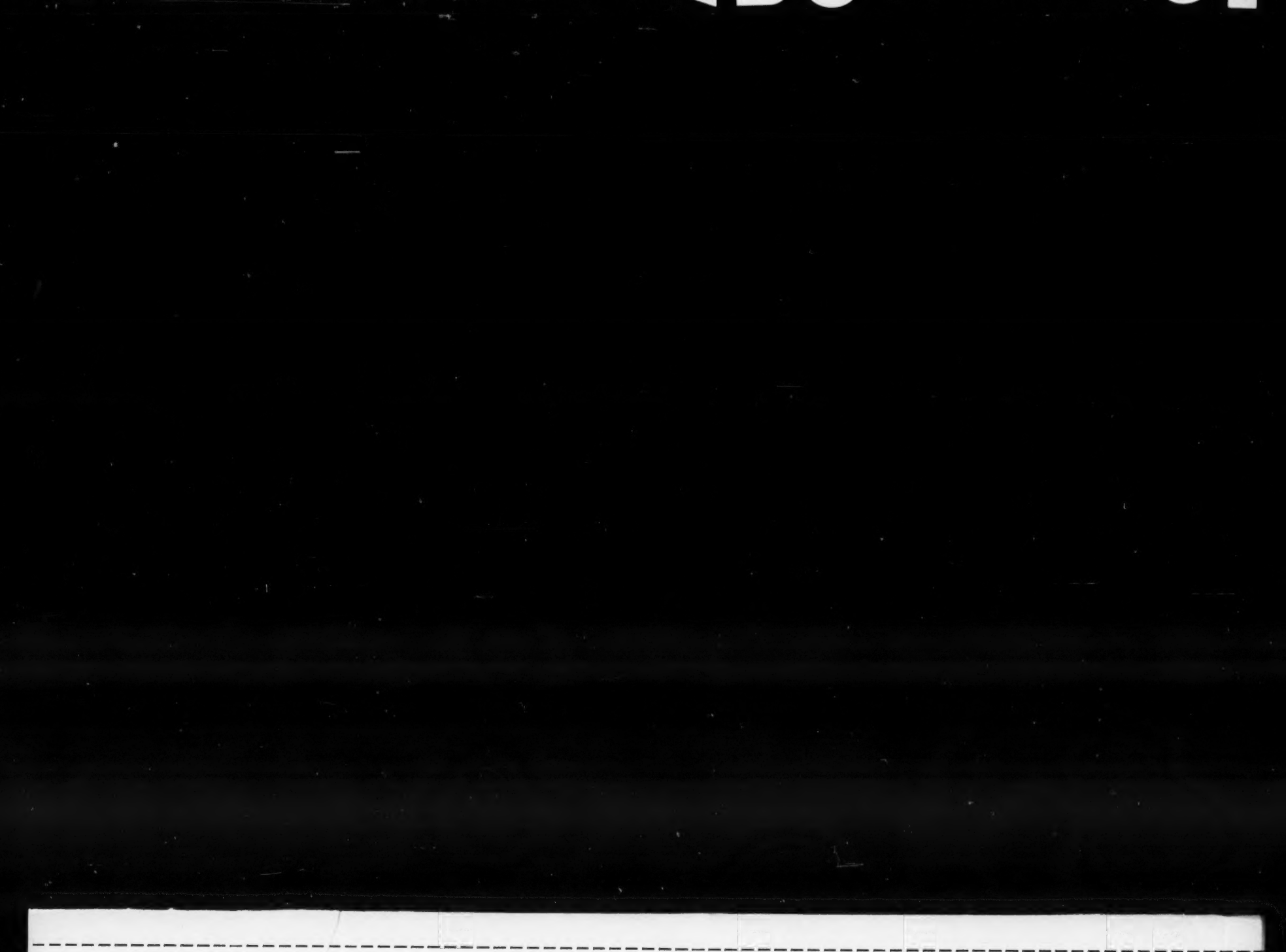
Commissioned and ex-commissioned officers of the regular Navy, Marine Corps and Coast Guard of the United States; warrant and ex-warrant officers of the regular Navy, Coast Guard and Marine Corps of the United States; reserve commissioned and warrant officers of the Navy, Coast Guard and Marine Corps of the United States shall be eligible as Naval Members. Persons eligible as naval members shall be admitted upon application and payment of annual dues.

Persons in civil life whose knowledge of engineering is such that they can cooperate with Naval engineers in the promotion of professional knowledge may be eligible as civil members. They shall have been in active practice of an engineering profession for at least eight years and in responsible charge of important work for five years, and shall be qualified to design as well as to direct engineering work. Fulfilling the duties of a professor of engineering who is in charge of a department in a college or school of accepted standing shall be taken as an equivalent to an equal number of years of active practice. Graduation from a school of engineering of recognized standing shall be considered as equivalent to two years of active practice. Persons eligible as civil members may be admitted upon application and payment of annual dues, provided that the application is accompanied by the recommendation of two members and provided that the application shall receive the approval of a majority of the Council.

Persons in civil life who are not eligible for civil membership, but who are especially interested in naval matters or the merchant marine may be eligible as associate members. Commissioned officers of the United States Army and of foreign military and naval services may be eligible as associate members. Persons eligible to associate membership may be admitted upon application and payment of annual dues, provided the application have the recommendation of a member and provided the application shall receive the approval of a majority of the Council, except that in the case of commissioned officers of the United States Army and of foreign naval and military services, the recommendation of a member will not be required.

Associate members shall be entitled to all the privileges of other members except voting and holding office.

The annual dues shall be $7.50, payable on 1 January in advance, of which $6.00 shall be for subscription to the JOURNAL of the American Society of Naval Engineers, Inc., for one year.



Secretaries of the

AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.

Captain J. E. Hamilton, U. S. Navy Retired, Secretary-Treasurer

Captain Robert B. Madden, U. S. N. Assistant Secretary-Treasurer

---

Past Secretaries

1889 P. A. Engineer *R. S. Griffin*, U. S. Navy
1890 Assistant Engineer *W. M. McFarland*, U. S. Navy
1891 Assistant Engineer *Emil Theiss*, U. S. Navy
1892-93 P. A. Engineer *W. M. McFarland*, U. S. Navy
1894-95 P. A. Engineer *R. S. Griffin*, U. S. Navy
1896-97 P. A. Engineer *F. C. Bieg*, U. S. Navy
1898 P. A. Engineer *W. M. McFarland*, U. S. Navy
1899 Chief Engineer *A. B. Willits*, U. S. Navy
1900 Lt. Cmdr. *A. B. Willits*, U. S. Navy
1901 Lieutenant *B. C. Bryan*, U. S. Navy
1902 Lieutenant *C. W. Dyson*, U. S. Navy
1903 Lt. Cmdr. *John R. Edwards*, U. S. Navy
1904 Lieutenant *M. E. Reed*, U. S. Navy
1905 Lieutenant *W. W. White*, U. S. Navy
1906 Lieutenant *C. K. Mallory*, U. S. Navy
1907-08 Lt. Comdr. *T. C. Fenton*, U. S. Navy
1909-10 Lieutenant **H. C. Dinger**, U. S. Navy
1911 Commander *U. T. Holmes*, U. S. Navy
1912 { Lieutenant *John Halligan*, U. S. Navy / Lt. Comdr. *E. L. Bennett*, U. S. Navy }
1913 Lieutenant **O. L. Cox**, U. S. Navy
1914 Lt .Comdr. **H. C. Dinger**, U. S. Navy
1915-16 Lieutenant *A. T. Church*, U. S. Navy
1917 { Lt. Comdr. **J. O. Richardson**, U. S. Navy / Lt. Comdr. *F. W. Sterling*, U. S. Navy }
1918 Lt. Comdr. *F. W. Sterling*, U. S. Navy
1919 { Lt. Comdr. *F. W. Sterling*, U. S. Navy / Commander **J. S. Evans**, U. S. Navy }
1920 Commander **J. S. Evans**, U. S. Navy
1921 { Commander **J. S. Evans**, U. S. Navy / Commander **S. M. Robinson**, U. S. Navy }
1922-23 Commander **S. M. Robinson**, U. S. Navy
1924-25 Commander **Bryson Bruce**, U. S. Navy
1926 Commander **A. M. Charlton**, U. S. Navy
1927 Commander **H. B. Hird**, U. S. Navy
1928 { Commander **H. B. Hird**, U. S. Navy / Captain **O. L. Cox**, U. S. Navy }
1929-30 Commander *H. T. Smith*, U. S. Navy
1931 Captain **O. L. Cox**, U. S. Navy
1932 Commander **H. F. D. Davis**, U. S. Navy
1933-34 Commander **H. B. Hird**, U. S. Navy
1935 Commander **C. S. Gillette**, U. S. Navy
1936 { Commander **C. S. Gillette**, U. S. Navy / Commander **Roger W .Paine**, U. S. Navy }
1937 Commander **Roger W. Paine**, U. S. Navy
1938 { Commander **Roger W. Paine**, U. S. Navy / Lt. Comdr. **Guy Chadwick**, U. S. Navy }
1939-40 Lt. Comdr. **Guy Chadwick**, U. S. Navy
1940-44 Captain **J. E. Hamilton**, U. S. Navy
1945 Commander **R. T. Sutherland, Jr.**, U. S. Navy
1945-48 Captain **F. W. Walton**, U. S. Navy
1948-51 Captain **J. E. Hamilton**, U. S. Navy

Past Assistant Secretaries

1939-40 Lieutenant **Milo B. Williams** (C.C.) U. S. Navy
1941-44 Lieutenant **Robert T. Sutherland**, U. S. Navy
1944 Captain **F. W. Walton**, U. S. Navy
1944-47 Commander **Floyd B. Shultz**, U. S. Navy
1947-49 Commander **C. H. Meigs**, U. S. Navy
1949-50 Commander **Frank C. Jones**, U. S. Navy
1951- Captain **Robert B. Madden**

Clerk and Administrative Assistant

1908- **Mr. Arthur G. Fessenden**

---

*Names in italics deceased